THE SOCIETY OF AUTOMOTIVE ENGINEERS

INC.

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TRANSACTIONS OF THE

SOCIETY OF AUTOMOTIVE ENGINEERS

INC.

PART II VOLUME XVII 1922

Papers Presented at the Semi-Annual Society and Section Meetings

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Society of Automotive Engineers, Inc.

Office of the Society
29 West 39th Street, New York City

The Society of Automotive Engineers
INC.

Founded 1905 — Incorporated 1909

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1910	Howard E Coffin
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1912	Henry F Donaldson (deceased)
1912	Herbert W Alden
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The Society shall not be responsible for statements or opinions advanced in papers or discussions at its meetings * * *
—(*Constitution, paragraph* 54)

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THE SEMI-ANNUAL MEETING

Seldom does one find so universal a feeling of genuine satisfaction among the attendants at a national convention as that which prevailed throughout the period of the Summer Meeting of the Society at White Sulphur Springs, W. Va., June 20 to 24, 1922. The attendance, though somewhat smaller than that of the last 2 or 3 years, reached 531 and all sections of the Country and branches of the industry were represented. An unusually large percentage of the members arrived early on Tuesday and remained throughout the entire meeting until Saturday afternoon. Every phase of the program was received with enthusiasm. The technical papers contributed much valuable engineering information and aroused a very active discussion.

STANDARDS COMMITTEE SESSION

The regular session of the Standards Committee, which convened at 10:30 in the morning of Tuesday, June 20, was attended by 128 members and guests. After brief introductory remarks, and the declaration of a quorum by President B. B. Bachman, Standards Committee Chairman E. A. Johnston called for the reports of the Divisions. There was also a report by the Lighting Division on the revision of the present S.A.E. Standard for Head-lamp Illumination.

The supplementary report on Motor-Truck Front-Axle Hubs was approved, with the supplementary table of ratings amended so that the former heading "Spindle load in lb. on solid-tire rating at ground" reads, "Assumptions on which calculations for spindle sizes were based." The reports on Ignition-Distributor Mountings, Magneto Mountings and Starting-Motor Flange Mountings were approved. The report on Breaker-Contacts was amended by omitting the reference to No. 8-40 thread size and approved. The report on Tractor Drawbar Adjustments as presented was approved.

The report on Flywheel Housings was approved as presented. The report on Crankcase Drain-Plugs was discussed at considerable length. In view of the expressed opinion that this subject has to do with design rather than standardization for interchangeability, it was referred back to the Engine Division for further consideration toward its being discontinued. The report on Motorcycle Carbureter Flanges was withdrawn by Vice-Chairman R. J. Broege, due to criticisms that had been received. The Engine Division is to reconsider this subject.

The report on Leaf-Spring Steel was approved as presented. The report on Steel Spring Wire was amended by omitting the reference to "round, cold-drawn wire up to 3/16-in. diameter, except for some types of springs used in clutches, which are hot-rolled," and omitting the word "helical" from the caption to the table.

The reports on Head-Lamps, Electric Incandescent Lamps, Electric Incandescent Lamp Voltages and Motorboat Lighting Voltages were approved as presented. The supplementary report on Automobile Electric Head-Lamp Lighting Specifications was discussed at considerable length. A motion to endorse the complete report of the Illuminating Engineering Society on the Rules Governing the Approval of Headlighting Devices for Motor Vehicles, dated February 1922, with a reference added as to the acceptance by one State of tests approved in another State, and omitting the paragraph under "Approval," which refers to "Tilting Devices," was lost.

The principal reasons given for opposing the recommendation of the Lighting Division were that portions of its report dealing with other than laboratory tests were non-technical, intended primarily for regulatory purposes, and were outside the function of the Society. The report of the Division, including Parts I and II, was finally approved.

The report on Aluminum Alloys was approved as presented. The report on Wrought Non-Ferrous Alloy Specifications Nos. 77, 78 and 82 were approved as presented, except for omitting from the captions mention of the purpose for which these specifications were formulated. It was felt that the latter information should be given in sub-captions or footnotes. Specification No. 83 was referred back to the Non-Ferrous Metals Division for joint consideration with the Electrical Equipment Division, in view of the work which is in progress in the latter on standard specifications for magnet wire. The report on White Bearing Metals was amended to include certain corrections in the percentages for Specifications Nos. 10, 10A, 11, 11A, 13 and 13A.

The report on Flywheel Pulley Lugs, as submitted by the Stationary Engine Division, was approved.

The Parts and Fitting Division's reports on Passenger-Car Front Bumpers, Rod-Ends, Plain Steel Washers, Ball-Studs, Serrated-Shaft Fittings, Tank and Radiator Caps and Lock Washers were approved as presented. The discussion on Rod-Ends developed the suggestion that the Division consider the extension of the standard to include a series of even heavier rod-ends for truck application, it being stated that the present standard sizes provide rather small pin-bearing lengths and diameters. The Screw-Threads Division's report on Screw-Threads was approved as presented. The report on Gages and Gaging, which was proposed for general information only, was referred back to the Division for further consideration in view of the criticism that it did not deal adequately with gaging for errors in lead.

The report on Top-Irons was approved after being amended to specify a ⅝-in. length of thread, and a 1-in. length of stud.

The Springs Division reports on Spring-Eye Bushings and Frame Brackets for Springs were approved as presented. The report on Definitions was referred back to the Division because of criticism of the method of defining deflection, load height and free height.

The Lubricants Division's progress report that was presented only to secure suggestions and information for the Division's guidance followed a meeting held by the oil producing and consuming interests during the morning. A number of valuable suggestions were received.

A progress report was made by the Chairmen of the Passenger-Car and the Engine Divisions on their study of methods of numbering engines and frames for theft prevention, and to secure reduction of automobile theft-insurance premiums. The report was supplemented by the exhibition of a number of models and by lantern slides illustrating the application of many methods that have been considered. The progress reports on Metric Thrust Ball-Bearings, Brake-Lining, and Starting and Lighting Equipment were not given on account of the lack of time.

R. M. Hudson, of the Division of Simplified Practice of the Department of Commerce, presented a very interesting paper on the work of that Division. He explained the service that it is felt the Department of Commerce can render the automotive industry and the public through organized cooperation with the Society of Automotive Engineers and the National Automobile Chamber of Commerce.

The action taken by the Standards Committee on the reports submitted by the Divisions was reported to and approved by the Council and at the Business Session of the Society held Tuesday evening.

NOMINATION OF 1923 OFFICERS

H. W. Alden was nominated to serve as President of the Society for the next calendar year by the Nominating Committee, which was completed and organized at the White Sulphur Springs Meeting. The committee reported the following other consenting nominees for the elective offices next falling vacant under the constitution, i.e., after the 1923 Annual Meeting of the Society:

First Vice-President—H. M. Crane
Second Vice-President, representing motor-car engineering—(Undecided)
Second Vice-President, representing tractor engineering—A. W. Scarratt
Second Vice-President, representing aeronautic engineering—E. P. Warner
Second Vice-President, representing marine engineering—E. J. Hall
Second Vice-President, representing stationary internal-combustion engineering—(Undecided)
Councilors (to serve during 1923 and 1924)—W. A. Chryst, F. W. Gurney and A. J. Scaife
Councilor (to serve during 1923)—H. M. Swetland
Treasurer—C. B. Whittelsey

The Nominating Committee was constituted of Cornelius T. Myers (chairman), Metropolitan Section; V. G. Apple, Dayton Section; H. R. Corse, Buffalo Section; T. F. Cullen, Pennsylvania Section; L. A. Emerson, Minneapolis Section; W. S. James, Washington Section; T. J. Lítle, Jr., Detroit Section; R. J. Nightingale, Cleveland Section; B. S. Pfeiffer (secretary), Mid-West Section; L. W. Rosenthal, New England Section; M. A. Smith, Indiana Section; and V. E. Clark, F. S. Duesenberg and G. E. Goddard, members-at-large.

BUSINESS SESSION

President Bachman's address was received very cordially at the Business Session held Tuesday evening.

It was reported that the Society's net loss for the first

8 months of the current fiscal year was $18,360.26. During the corresponding period of the last fiscal year the Society had an unexpended income of $14,504.88. The loss in income this year is due to a reduction in receipts of $27,439.30 as compared with last year. The amount of initiation fees from new members was $6,335.00 less than for the same 8 months of the last fiscal year. As a result of careful management the total operating expense for the period was increased only $5,425.84, notwithstanding added activities involving expenditure of approximately $10,000.

On April 30 the assets of the Society amounted to $180,127.63, these being offset by accounts payable of $9,159.29 and special reserves of $50,758.11; leaving net assets of $120,210.23, approximately $90,000 of this amount being in the form of United States Government and railroad securities.

The Membership Committee reported that the total enrollment of the Society on May 31, 1922, was 132 more than on the corresponding date of 1921, notwithstanding the fact that several hundred members were dropped for non-payment of dues or other causes.

The Sections Committee announced that all of the Sections were in a healthy financial condition and that many excellent papers had been presented and discussed at their meetings during last season. The committee advised strongly the practice of the Sections arranging their programs for the year in accordance with pre-determined plans.

Under the item of new business, a lengthy discussion was had on the Society's affairs in general, including the matter of the grading of applicants for membership, Sections activities and the holding of local meetings in cities at which no Sections of the Society are located.

HIGHWAY MATTERS

Director W. K. Hatt, of the Advisory Board on Highway Research of the National Research Council, presented at the Friday morning technical session a valuable uptodate survey of studies on highway matters currently conducted by different institutions throughout the Country. Professor Hatt later conferred with H. W. Alden, chairman, G. A. Green and Prof. W. E. Lay, of the Highways Committee of the Society, with a view to coordinating further the technical efforts of highway and automotive engineers.

PRESIDENTIAL ADDRESS OF B B BACHMAN

The activities of the Society during the period since we met in New York City have been numerous. I would like to lay a few of them before you in the way of a report, covering in general the problems that have come before the Council. This can be accomplished best by following the work as divided among the administrative committees.

THE MEETINGS COMMITTEE

The results of the activities of the Meetings Committee during this period need little comment as the material evidences are before you, and will unfold during the session in which we are now participating. On the basis of the work that I know they have each done individually, I believe that I can anticipate your unanimous approval, and extend to the Committee your appreciation as well as that of the Council for the sacrifice of time and effort that they have made.

For the future, I know the Committee is interested in the activities of the Sections, of which I will speak more in detail later, and will welcome the opportunity for more active cooperation with the Sections in establishing a rounded-out and harmonious program of meetings for the year. As a large part of the value of the Society to its members lies in the meetings that are held, there has been some discussion as to whether it would not be wise to hold annually more than the two meetings for which the Constitution provides, and very definite thought is being given to this subject.

SOCIETY MEMBERSHIP

The Membership Committee has been dealing with a very serious problem in the affairs of the Society. The growing activities and the new fields of work that are opening around us on every side require the addition to our ranks of all who are equipped to assist us in our work or who can be benefited by it. While the scope of our organization is large, it is in a certain sense, limited. We must keep in mind that restricting our membership to the field of active designers of automotive vehicles and their more important component parts would be a very narrow policy. On the other hand, we must recognize that opening our doors for the general admission of all without regard to the service we can render them or they can render us would be unwise. Between these two extremes, the Committee and your Council believe, there is a sufficiently large field for the Society to draw from in permitting a rational and satisfactory growth.

This field, it seems to me, will cover, first, the designer of automotive apparatus and their important component parts; the instructor in arts and sciences relating thereto; the specialist, or the man versed in the design, construction and efficient operation of the production agencies for materials and completed structures; and, finally, but by no means least, the man who is skilled and is competent by training and experience in the maintenance and operation of the apparatus that the industry manufactures. This view may be criticized as being too broad, but I am firmly convinced that unless we recognize the valuable assistance that these other classes can render in developing the field of internal-combustion power application, and are willing to receive them on the basis of equality to which their ability and dignity entitles them, we will be retarding our attainment to that position of consideration to which we are entitled.

In the class of those who, while not engineers in the broad sense outlined above, are nevertheless interested in the work that we are doing, and directly or indirectly are benefited by it, there should be included with the man engaged in marketing our products, the man who has the responsibility of purchasing materials that we use in our processes of manufacture. Both these classes can in many instances give a wider perspective to our vision and can be materially benefited by our activities.

While the continuing growth and activity of the Society require careful consideration of our financial resources, and while the fees that are received from our members are a very important source of revenue, we must guard against any plan of membership increase that has only revenue in mind. The work that the Society is doing and the service that we can render to each other as members cannot be measured by the cost, nor should we knowingly solicit membership for mere financial support from anyone.

Another problem that the Membership Committee has actively in mind is the question of retaining the active interest of those who are already members. The recent industrial conditions have brought about the reduction of organizations which has resulted in personal difficulty for some of our members. There are others who possibly have been persuaded to join on the wrong basis, and have therefore never really been members.

In any event, whatever the reason may be, it is probably only natural that there should be a percentage who are delinquent in their financial obligations and thereby, as a result of our constitutional provisions, deprived of some of the benefits of membership. Extraordinary efforts have been made during the past months to increase the value and effectiveness of our employment service for the assistance and benefit of those who are numbered among those who have suffered personal loss through the recent unsettled conditions. As for the others, it is difficult to say just what steps should be taken to stimulate their interest and to retain their active association with us, but several methods are being actively canvassed toward this end by various committees and the Council.

The Sections

The work of the Sections Committee in the phase of the Society's activities that it represents is undoubtedly of prime importance. As was to be expected, the proper organization of these activities has presented for a number of years some very intricate problems which are far from being settled, even today. Fundamentally, an organization of the character of ours is dependent in a large measure upon two things for its success; first, the character and value of its meetings; and second, the character and value of its publications.

It is manifestly impossible with a widespread membership that more than a relatively small percentage can attend meetings such as this and at the same time the multiplication of meetings on a national scale at more frequent intervals, while desirable in some ways, will not in itself fulfill all the possible functions of an active local Section. The theory upon which we have conducted the affairs of the Sections to date has been to give them a very large measure of independence, allowing them to direct and regulate their affairs in accordance with their local needs and under officers of their own selection.

They have been financed in large part by the contributions of the members of the Society who have joined these Sections, the Society contributing to the support of the Sections in addition. The reason that the present system has been adopted is that it has been felt that only a percentage of the membership of the Society would be so located geographically as to benefit by participating in Section activities, except to the extent that the multiplication of Section meetings with the presentation of valuable papers would furnish subject matter for THE JOURNAL. In view of this, it was felt that, even if it were possible, it would be wrong to appropriate from the funds of the Society the necessary amount to finance completely the Sections whose activities would be of direct benefit to only a portion of the membership. At the same time, on the basis of the increased contributions to THE JOURNAL, it was felt that some appropriation was justifiable.

Against this theory of the present method, we have the view that those members who wish to participate in Section activities should not have to pay further dues than those to which they obligate themselves in joining the Society; and it is urged that this additional taxation is in a large measure responsible for difficulty in bringing up the membership of the Sections to what it should be. There is much to be said for both of these viewpoints, but fundamentally I do not believe that in either one of them lies the secret of success or the reason for failure in the Section activities.

It has been suggested that a very large percentage of our members do not care to attend Section meetings, and unfortunately this is probably only too true; but I believe that one of the reasons for this lack of willingness to attend is that a proper survey of the needs of the members has not been made and the programs that have been put forward have not been consistently of the caliber or kind to attract a consistent attendance. This statement is not made with any intention of criticizing any of the past or present Section administrations. It is merely a fact that I believe we must face thoroughly, and a problem that we must solve before we can put the Section work on the high plane where it should be.

In a large degree the Section problem is similar to the Society problem. While we recognize the sacrifice of time and the effort and thought contributed by committee members, we surely recognize that if it were not for our headquarters organization we would be lost. I know of no individual who has the ability for organization and the time to devote to the detail work of properly conducting a Section. The result is lack of continuity of effort and policy.

SOCIETY FINANCES

During the last year, the question of finance has caused your officers considerable thought and anxiety. For the first time in many years, our current revenues have been insufficient to meet our expenses and afford a margin to be transferred to surplus. This is due to several things: first, to a loss in income from our advertising; and, second, to a reduction in the number of new members. Both of these conditions, it is believed, are temporary and will show improvement with a recovery of normal business conditions. We should nevertheless recognize that growth in usefulness and numbers will require not only continuation but expansion in our services, which will require thought and careful planning to balance the budget. It would have been possible this year to meet this emergency by a reduction in our activities, and it would be easy to recommend raising dues to prepare for the future. It is, however, the feeling of your officers that it is in times of commercial difficulty that organizations of the character of ours should increase rather than decrease their activities, for the reason that it is during times of this kind that the members individually and the industry as a whole need the greatest stimulus.

Regarding the raising of membership dues, there are of course many who could meet an increase with little difficulty, but on the other hand there are a number who, while they are vitally interested in the work of the Society, have found it impossible to meet the current dues. It has been my privilege to conduct a rather extensive correspondence with this smaller group. This has been, of course, in many instances of a confidential nature, but I am not violating that confidence in telling you that the opinion expressed above has resulted from this contact.

Nevertheless, we had to meet this problem. A survey indicated that we were spending a considerable amount of money annually in publications. While it is recognized that this is a legitimate and important function of the Society, conditions have changed; and the changes had not been reflected in the publications' policies. When the Society was first organized, it, in common with other engineering organizations, published its proceedings in the form of an annual or semi-annual volume containing complete papers and discussion of them. So long as there were no other or better avenues for the distribution of information to the members, this was very well. However, in later years, we have, through the activities of the Standards Committee, published the S.A.E. HANDBOOK; later the *Bulletin*, which has developed into THE JOURNAL, was brought into existence. In THE JOURNAL we have a means of presenting to the members at a much earlier date than was possible in the TRANSACTIONS a complete record of the proceedings of the Society. Therefore, it seems that it would be highly inefficient for the Society to continue indefinitely to distribute the complete proceedings in THE JOURNAL and then at a later period duplicate this information in the form of a bound volume, without charge to the members in addition to dues.

There are many questions connected with this problem which it would be impossible to cover except in an inexcusably lengthy manner. Suffice it to say that, after long and careful consideration, the Council has finally decided that the TRANSACTIONS for the years 1921 and 1922 shall be sent to only those members who indicate that they wish to receive them. There will be no additional charge for these. After that time, it is the recommendation of your present Council, that the TRANSACTIONS be sold to the members at a nominal price which will partly cover the cost of production. This will permit several things: first, it will enable us to concentrate in greater degree on making THE JOURNAL more uptodate and complete in its record; and, second, it will relieve the finances of the Society of a burden that in a large degree under the old order was imposed for a service that was of little or no benefit to a large proportion of our membership. That this viewpoint is correct we believe is demonstrated by the fact that there were orders for only about 1200 copies of the last issue of the TRANSACTIONS.

I can appreciate thoroughly that a change of this nature will seem radical to some. I also appreciate the powerful influence of precedent and the fact that the receipt of bound volumes of TRANSACTIONS has long been a perquisite of members of engineering societies. On the other hand, this Society has in a large degree established itself and justified its existence as an organization on the basis of a disregard of precedent; and I believe the other arguments that I have outlined herein are ample justification for the step that your Council has taken, and I trust that the development of the plan will recommend itself to those of our loyal and interested members who have felt inclined to question the wisdom of the step.

The Standards Work

With regard to the Standards Committee, I think you will recognize that, in view of my long association with this phase of our work, I am most vitally interested in what is being done. We were fortunate this year in being able to get a complete working organization of the Standards Committee going very promptly. A considerable amount of work has been done, as evidenced by the reports that were presented to the whole Committee by the Divisions this morning. This part of the work speaks for itself, and I will not do more than make this reference to it.

There are, however, certain other phases of the Standards work to which I wish to call your attention. At the risk of being tiresome, I would repeat what has been said so often before, that the Standards work is one of the most important activities of the Society. It has been suggested that the direct benefits of this work have reacted in favor of the industry as a whole, rather than of the individuals who hold membership in the Society and from whom we obtain financial support in large measure, and that, in view of this fact, it would be well if means could be found that would place the financial burden for the support of this work on the shoulders of those who most largely benefit from it. While there is no doubt as to the soundness of these suggestions from the viewpoint of placing the burden of expense on the shoulders of those to whom the benefits accrue, there are other considerations which should have our thoughtful attention. I am placing before you herein what are largely my own opinions, and hope that you will recognize them as such.

I believe that the fundamental strength of the Society resides in its being an association of individuals, and that the strength of our position as an impartial agency for the conduct of many of our activities would be jeopardized were we to make provision for corporate memberships the main purpose of which would be to obtain revenue. We have been fortunate in having our work recognized by several trade organizations which have indicated their approval and support by making financial contributions. The degree in which these have come to us, and representing as they do, not individuals but groups, I believe is therefore the most practical solution for this problem. It would be of considerable assistance and I believe perfectly proper if this form of recognition were extended. However, whether it shall be or not, we should exercise all our ability and energy to proceed in a rational way to extend and continue the Standards work that was inaugurated about 12 years ago, and has been carried on continually since. Cooperative endeavor of this sort, bringing together as it does the individual members of the Divisions, is of the greatest benefit in promoting the development of the individual and the building-up of the spirit of service that is vitally essential to the health and growth of such an organization as ours.

I wish that we could find some way of still further impressing upon the industry the importance of this work; that we could find a successful method of definitely determining the degree to which the S.A.E. Standards are used and a more definite measure of the economies that their use brings about. While it is necessary for the purposes of efficient organization that the Divisions be not too large, I would like to see the time arrive when the Division meetings should partake of the nature of technical sessions to which not only the Division members but all interested members would feel free to come and would desire to come. One particular reason for this feeling is my belief that as we grow and continue this work we must be more and more particular that the subjects proposed for standardization are properly considered and thoroughly analyzed in view of the broadest possible experience and opinion, so that assurance may be had that all interests have been properly represented. I wish that we could individually make it our plan and purpose to sell the idea that it is good business and money well invested for organizations to give their engineers the time and to assume their expenses in the attendance at these meetings as well as those of a more general nature.

As we proceed with the work of standardization, we will encounter more and more difference of opinion as to how far it should be carried. There are those among us whose breadth of vision carries them far in the list of subjects that they believe can rationally be standardized. There are others who feel that these suggestions if followed would be unwise, and would result in harmful restriction of initiative in the design and development of our apparatus. I hope that these two views will always be in evidence, but that they will be brought into contact in the work of the Committee, so that they may temper each other and produce a rational result. I believe that a student cannot fail to recognize that it is in the reaction of the extremes of opinion in their contact with each other that sound and conservative policies are formulated.

In connection with the work of the Standards Committee, it is well for us to recognize the increasing recognition of the importance of such work in every quarter. In the January issue of the *Automobile Engineer* Basil H. Joy outlined the work of seven committees, operating under the British Engineering Standards Association, having to do with the general subject of automobiles. These subcommittees are dealing with nomenclature, steel, small fittings, electrical fittings, wheels, rims and tires, and cast iron. You will recognize that for practically all of these our Standards Committee has already brought into existence valuable standards. The Department of Commerce, under Secretary Hoover, in the Division of Simplified Practice, is taking a very active interest, as the name of the Division suggests, in the simplification that can be obtained by the adoption and use of standards. You have had the opportunity of hearing today from Mr. Hudson of the Division exactly what its aims and ideals are, and we hope in cooperation with the representatives of the National Automobile Chamber of Commerce to be able to further this work.

RESEARCH

With regard to the work of the Research Committee, I feel that it would be presumptuous for me to attempt to make an extensive statement. We are devoting to this subject a session which, in conjunction with the report of Mr. Crane, chairman, and Dr. Dickinson, manager of the Department, will go farther than it would be possible for me to go, in outlining what has been done and what it is proposed to do.

My remarks on the work of the Standards Committee bear with equal force on the work of the Research Committee. The work in itself is largely of a character that will produce benefits that, in a considerable degree at least, will permit their being secured by others than the individuals who comprise the membership of the Society. This viewpoint should not, however, blind us to the potential benefits that can accrue to the members. I say "potential" for the reason that the benefits will not be secured except insofar as the membership participates in the work and what each member will get out of it will, in a large degree, depend upon what he puts into it.

The results may not be startling in their scope at the present time, but I am firm in my conviction that the foundations that have been laid, if built upon with patience and with the consistent support of the membership, will in the very near future justify the inclusion of this work as a part of our regular program.

NEW DEVELOPMENTS

After this more or less hurried summary of the affairs of the Society, I would direct your attention to a more general survey, with a view to determining along what lines our activities as engineers and as an engineering society should be directed in the immediate future.

The period of industrial depression through which we have gone should be productive of some lessons to which it would be well for us to give thought. Naturally, those that appeal to me most forcibly and which I feel most competent to discuss are those having to do with the truck rather than the passenger vehicle. There have been three outstanding developments during recent months, the appearance of which may be due in part to conditions resulting from the depression. They are: the speed-wagon, the motorbus and the motor rail-car. That there is a fertile field of usefulness for all three of these types can probably be accepted without question. That they each present features of design requirements which are distinctive and possibly not yet fairly appreciated in general is, I believe, also true.

We held in January and will hold at this meeting a session dealing in a degree with the problem of bus transportation. There have been sessions held by the Metropolitan and the Indiana Sections that had to do with the matter of the motor rail-car. The problem of the speed-wagon may be more commercial than technical, but I believe that it deserves consideration. I mention these points with the hope that our Sections will find some suggestions for their development as meeting topics.

HIGHWAYS

The question of highways is one that has been given considerable attention in the past in our discussions and should receive continuing attention. The ability and the efficiency of the vehicles that we construct are dependent in a large degree upon the character of the roads upon which they are operated. While it is true that the invention and development of the automobile has increased the demand for improved roads, it is also true that the growth of improved roads has increased the demand for and use of the motor vehicle, and future limitation in road construction will act as a limitation on the vehicle market.

It appears to me to be particularly unfortunate that there should be any controversy between the railroads and the users and builders of motor vehicles, instead of complete harmony and cooperation. Except in the most isolated cases, competition between these two forms of transportation is most unlikely. I think this is almost universally true with regard to transportation of goods; and in the transportation of passengers it is almost equally true if we stretch our imagination to embrace what must be the development of the future. I recognize the fact that there is a large amount of capital invested in street-railway transportation, but I am also impressed more and more daily with the fact that the streets of our cities are becoming less able to accommodate the burden of traffic that they are called upon to bear. It seems to me not at all improbable that this condition will make it imperative in the not very distant future to replace track vehicles with a more flexible form of vehicle for short hauls and where frequent stops are necessary.

This problem of highway capacity as evidenced by our city streets deserves the most careful study on the part of every automotive engineer, particularly as to what its probable effect will be on future design requirements as affecting the size of the vehicle, the control with respect to steering, turning-radius, acceleration and braking. In many of our cities very stringent regulations with regard to parking have been put into force. It is useless to spend our time in railing against these provisions, for in some measure at least they represent the legitimate effort to distribute the use of the streets in a fair way among all citizens. The problem presented is of the most complex nature and deserves careful study and analysis.

Another result of the increasing traffic-density is the lowering of the efficiency of motor vehicles as a means of saving time. As the cost of operation of motor vehicles has been reduced, and the possibility of use thereby increased, this new factor of limitation of speed, due to congestion, becomes increasingly important.

In the broader aspect of transportation in rural and suburban communities there should be practically no question of conflict between the railroad and the motor vehicle. We have in this Country a sufficiently close-up picture of the development of transportation facilities to be able to get a very comprehensive and intelligent view of the relation between various means of transportation and the establishment and development of communities.

The early settlements were along the seaboard and the more navigable streams, and this condition of affairs continued up to the time of the development of the railroad, which resulted in the unlocking of the vast inland empire and the linking-up of the Pacific coast with the Atlantic, which would have been practically impossible without this new means of transportation. The development of electricity and its application to high speed interurban lines was the next step in bringing high-speed transportation into closer contact with the small community and individual. It is obvious, however, that the operation of rail lines calls for a virtual monopoly of territory in the form of a franchise, and limits the operation of vehicles over any given track to one centralized authority, and calls for fixed schedules of operation.

The advent of the automobile has resulted in placing into the hands of the individual a smaller and more flexible unit with practically the equivalent speed-capacity of the railroad. This vehicle, capable of being operated over the road, can be made more truly competitive and infinitely more flexible and independent of fixed schedules. The growing use of the automobile and the truck, coincident with the development of and as an auxiliary to the railway system, has resulted in extensive suburban and rural development which would probably have been as impossible without the automobile as the development of the inland cities of this Country would have been without the railroad.

While this development has resulted, and the increase in realty value is recognized and acknowledged, the increasing traffic, particularly over main routes, will bring a reaction unless we are peculiarly alert to study and suppress in design all objectionable characteristics of our vehicles to the greatest possible degree. I appreciate that the control of all these features is not in the hands of the engineer or builder, but he should be thoroughly posted as to what they are and be prepared to cooperate intelligently with regulatory bodies to assure that rational measures for the protection of the public, which do not impose unreasonable restriction on road transportation, are enforced.

Originally road construction was in the hands of individuals or corporations that operated them for profit in the collection of tolls. While we have rejected, as a Nation, the idea of public ownership of the railroads, so also have we rejected the idea of private ownership of the highways. I believe both these ideas are proper. In the railroad we require concentration of authority and responsibility in operation over any one given line. This can be obtained most efficiently by private ownership and operation under reasonable government regulation. The highway, on the other hand, is primarily for the use of the individual according to his needs and desires, with as little restriction as possible consistent with public safety; this can be obtained best by public ownership and complete government control through one of its departments.

Much of the discussion on the question as to who should bear the burden of the cost of construction and maintenance of our highway systems, or whether the motor-vehicle operator is receiving a public subsidy that is not shared by the railroad, etc., appears to be beside the point. The cost of transportation of passengers and freight, by railroad, water or highway, is borne by the whole community and shared by every citizen in proportion to his requirements for transportation. I believe this to be so, whether the cost of transportation is included in the cost of the commodity or it appears partly in the form of taxes. The big fundamental problem is to determine the economic field for each medium of transportation and the relation each should bear to the other for maximum efficiency, and the most satisfactory means of proportioning the expense to the individual.

I have endeavored to the best of my ability to give you a brief and yet comprehensive view of the problems that are confronting us and should receive our active individual and collective attention. I hope the result will be to stimulate interest in the affairs of the Society and to enlarge our view of the future activity of, and the service that can be rendered by, each of us individually and all of us as an organization.

PROGRESS OF THE RESEARCH DEPARTMENT[1]

By Dr H C Dickinson[2]

Dr. Dickinson outlines the history of the Research Department since its organization, indicates why the universities are the principal bases of operation for pure research, describes how the department functions as a clearing-house with regard to research data and comments upon the bright prospects for the future. He enumerates also the facilities the Research Department has for the coordination of research problems.

The practical achievements of the Department have resulted from its recent concentration upon the three major projects of study with regard to the tractive resistance of roads, with reference to fuel and to testing programs, and of an effort to render financial assistance to the Bureau of Standards and the Bureau of Mines that would enable these Bureaus to continue their elaborate research programs, details of all of this work being included.

Supplementary road tests now being conducted by nine different automotive-vehicle companies are outlined, and the factors governing the selection of fuels for test purposes are stated and commented upon.

Six months ago, at the Annual Meeting of the Society in New York City, the work of the Department was outlined by H. M. Crane, as Chairman of the Research Committee. The Department was then 4 months old. Its program had been mapped out but little practical work had been done. The last 6 months have gone to show what the Research Department can do for the industry and how.

The automotive industry is one of the three largest in the United States. It is generally admitted that the American designer is far ahead of his foreign competitors. America created the industry and still has the lead. It is an industry composed of talented young men, remarkable alike for adaptability and initiative, with no traditions or precedents to hamper their progress. It is intimately bound up with, and dependent upon, a number of other great industries, being a very large consumer of iron, steel, gasoline and lubricants, rubber, paint, aluminum and all manner of accessories. If the source of supply of any of these materials fails, the automotive industry suffers a setback. During the war, when supplies of rubber, shellac and mica were controlled, and other commodities vital to our industry were hard to get, we realized, probably for the first time, our dependency upon other industries, and our general unpreparedness for abnormal conditions. Since that time, the engineers and manufacturers have taken a wider view. Realizing that a shortage of any one commodity may paralyze business, they engage research men to investigate the possibility of replacing that commodity by some other more plentiful material that will serve the purpose as well or better. Again, the automobile builder, anticipating a possible shortage of petroleum products, sets his laboratory men to work on the investigation of other fuels to be used in place of gasoline, and, if he is very foresighted, sets his engineers to designing engines that will burn substitute fuels efficiently and economically.

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[1] Semi-Annual Meeting paper.

[2] M.S.A.E.—Research manager, Society of Automotive Engineers, Inc., New York City.

The two problems I have just cited are problems affecting the industry as a whole. In addition to such problems as these, every manufacturer has his own particular problems, arising from his desire to make his product give better service than that of his nearest rival. He is constantly on the alert to develop new qualities that will make his car stand out as a good car. Any progressive manufacturer knows how necessary technical research is, if he is to make a success of his business. It is gratifying to note how much interest the manufacturer and engineer are taking in pure research, by which I mean the conduct of investigations not so much for the purpose of obtaining practical and profitable results, as for the advancement of knowledge in one particular problem affecting the industry as a whole. But, unfortunately, very few companies are blessed with a vision of the future combined with a money surplus; the rest cannot afford to indulge in pure research, which offers no prospect of an immediate monetary return.

Pure Research in Universities

The principal bases of operations for pure research are the universities. The university instructors and research men are actuated rather by a love of knowledge for its own sake than by any pecuniary gain that may accrue. Their pursuit of knowledge is an end in itself and not a means to an end. The results obtained by university laboratories are very wide in range and inestimable in value. While the main object of the universities always must be instruction, the presence of active and enthusiastic research groups is an ideal, if not an essential, background for the training of young engineers. Many of the schools are suffering from a great handicap. Owing to the scarcity of good teachers and the large number of men seeking instruction of a more or less elementary nature, many of the best potential training-grounds for research men tend to degenerate into mere technical schools imparting textbook information instead of principles, and giving too little individual attention to the students. Under such conditions research languishes, and the teacher loses his morale and becomes a teaching drudge instead of an intellectual leader. I cannot impress upon you too strongly the necessity for a constant supply of competent research engineers, well grounded in principles, to carry on the work of research in the industry. There can be no progress unless we have such men. It is to the universities that we must look for them. I would recommend that before the close of the school year, every man in the industry who employs young research engineers send a list of his requirements to a selected list of universities, so that he may have an opportunity of finding the type of man he needs. I have received a number of letters from young men, most of them completing post-graduate courses, keen research men with ability and enthusiasm, all wanting to know how they are to find an opportunity to turn their training to advantage. One of the larger British universities has an Appointments Board composed of professors and business men, whose task it is to keep in close touch with the industries and find out what positions are open for trained technical graduates. Any young man about to graduate is at liberty to go before the Board and state his case. Since practically every one of the large industries over there has established its research organization, the trained man is in demand, and none of the fine research material that the University produces is wasted.

A Clearing-House

Turning from the worker to the more pertinent question of the work itself, a review of the research field revealed a number of organizations, both schools and manufacturing laboratories, all engaged in interesting investigations of their own without any relation to one another. Every day valuable results were being obtained and only a few individuals were getting the benefit of them. Manufacturers were looking vainly for information that was available, but they did not know where to go for it. Over and over again work was being duplicated because the people doing the work had no idea of what other people were doing, while more important and pressing problems were awaiting solution. Everywhere the need of organization and of centralization was apparent.

The first task of the Research Department was to establish a clearing-house of information for the benefit of the industry. Soon after the department was organized, 2 weeks were spent in visiting industrial and school laboratories. The results of these visits were very illuminating. Everywhere we found engineers, professors and students ready and eager to discuss their problems, enthusiastic over the prospect of exchanging information with other people, people who had common research interests, and anxious to coöperate with the Department in every possible way. We had a large index-card that was described in THE JOURNAL for December 1921, which we asked them to fill out for our files, so that we might know exactly what work they had done, what they were doing, and what information they needed. In some cases we were able to supply them with the information they required at once, from the material we had on file. They were glad to hear of the organization of a central body to which their problems could be referred, and to which they could apply for information as to what was being done by other laboratories in their own field. They all agreed that, while a certain amount of duplication was necessary and even desirable, the Research Department could serve a useful purpose in preventing unnecessary duplication. Almost every laboratory suggested problems that called for solution, and that could not be undertaken by them owing to scarcity of funds, or of time, or of personnel. Some of the laboratories asked for suggestions as to what problems could be undertaken advantageously and were provided with a list of some of the problems in which the industry is interested. There are on file with the Department a number of suggestions for problems to be undertaken in the future; questions that are of importance but not of immediate urgency. In most cases the report of the laboratories was the same, "the harvest indeed is great, but the laborers are few."

The manufacturers, many of them, were suffering from the slump and had cut down to the bone. Many had dispensed with their research staffs, and the rest had had to retrench. There were notable exceptions, which will occur to all of you.

THE FUTURE

The prospects for the future are bright. The industry has come to realize the necessity for research and will soon be putting their principles into practice. More valuable still, as indicating the confidence that the Department has already inspired, engineers throughout the Country have developed the habit of taking the Department into their confidence, of writing as a matter of course describing what they are doing, and inviting suggestions as to the method of attack and the general conduct of their researches.

Recently another trip was made to some of the laboratories of the Middle West, and the progress was noticeable. The interest that the Research Department was taking in organizing research work had proved a real inspiration and the results obtained throughout the trip were most gratifying. We hope that eventually all the school and university laboratories engaged in automotive research will be visited, but it will be some time before trips to the Far West can be undertaken. Although the benefits of this field work, in bringing home to members the help they can expect from the Department, have been clearly demonstrated, it is doubtful whether the industry is aware of the special facilities which the Department has to offer.

RESEARCH DEPARTMENT FACILITIES

The Research Department is in close touch with the Engineering Societies Library, which is located on the thirteenth floor of the building in which the Society is housed. It is purely a reference library and contains most of the material that has been published on the automotive industry since its inception. The proceedings and journals of the various engineering societies, American and foreign, are kept on file, as are over 1000 current periodicals along engineering lines. For the benefit of engineers who cannot use the library themselves, the librarian maintains a photostat service, by which a complete copy of a magazine article referred to in another periodical may be had at a nominal cost. Bibliographies, references and translations may be had quickly, reliably and at the minimum expense. The fact that the great resources of this library are constantly at our disposal increases the efficiency of our Department enormously. If any one of our members wishes to know what has been published on the subject of brake-linings, or of pistons, for example, we can supplement the information we have in our own files by referring to the publications on the subject in the library. As a general rule, when we receive a query of this kind, we give a short bibliography of the subject, adding particulars of where the various publications may be obtained. For our own use, we have the Engineering Index, the Industrial Arts Index, and Science Abstracts, which list recent articles by subjects. This we supplement by our own indices, one covering the publications of other organizations and individuals, and one covering the publications of the Society. These indices are kept by authors, titles, and subjects, and involve a careful scrutiny of all the periodicals, American and foreign, in the automotive and allied fields. In connection with the first-named index, we have a file of pamphlets and publications that are not allowed out of the Department, since they are constantly needed for reference. In the Members' Room of the Society, there is a small library in which copies of current magazines in the automobile, aviation and tractor fields are kept up-to-date for the use of members, as well as THE JOURNAL of the Society, *The Engineer*, *Engineering*, and a number of other very valuable publications. These volumes are kept for reference and are available at all times.

Inquiries of all kinds are received from engineers, arising out of everyday experience and covering a variety of problems in automotive engineering. Incidentally, we have received two or three communications from Cuba and the Canal Zone, asking for information as to what adjustments should be made in an American car to enable it to run on alcohol, the only fuel obtainable at a low price in those territories. Reviewing the variety of problems that are presented to us for solution, we cannot fail to realize that the scope of the work is almost unlimited. Every effort is made to keep in close personal touch with engineers throughout the Country, who are engaged or interested in pioneer work. Many of them have developed a habit of dropping in at the Society's headquarters whenever they are in New York City, to talk over their problems.

In each issue of THE JOURNAL the Research Department discusses some topic that is of interest to the automotive industry, and suggests various problems for research arising from an examination of the subject. The discussion is always accompanied by a selected bibliography. The Department plans to publish in future issues of THE JOURNAL short abstracts of books and articles published here or abroad, which seem to be of interest to the industry, so that members may be sure of being informed of what is being published on their subject.

I have with me a supply of index-cards that were designed to afford ready reference to the activities and the questions of those firms and individuals who are interested in research, and I shall be glad to give one to anybody who is interested in cooperating with us and in getting the full benefit of the service we have to offer.

To pass from the resources of the Department to its practical achievements, we have been engaged for the past few months upon three major projects. The first of these is the cooperation with the Advisory Board on Highway Research of the National Research Council in the study of the tractive resistance of roads. With the advent of the commercial motor-vehicle as an essential part of our transportation system, and with the widespread use of passenger cars for purely commercial purposes, the problem of highways emerged from the "good roads" stage to become one of the most urgent of our national problems in industrial development.

While the problem may appear to be primarily the concern of the economist and the highway engineer, it is equally one for the automotive industry. The Department was originally drawn into this field through a request for cooperation with the Bureau of Public Roads and the National Research Council. A survey of the situation showed that there are many phases of the highway problem in which the cooperation of the Society is necessary to assure full consideration of the interests of the industry, as well as the benefit of its technical experience.

Recognition of this situation has led to the recent appointment of a Highways Committee, of which H. W. Alden is chairman. This will assure the needed contact between the Society and the other agencies engaged in highway research.

In addition to the foregoing activities, we have taken an active part in securing materials and financial support for a project which the Advisory Board on Highway Research has under way at the Massachusetts Institute of Technology under the direction of Major Mark L. Ireland. Extensive road tests have been made and valuable data are now in process of compilation.

FUEL RESEARCH

One of the major activities of the Research Department has been providing for the fuel-research program in progress at the Bureau of Standards. The two other main research activities of the Department have to do with the fuel problem. While many of you are familiar with the history of this undertaking, I shall explain it briefly for the benefit of those who are not. About a year and a half ago, a conference was called at the instance of Dr. Manning, director of research for the American Petroleum Institute, to discuss the joint responsibility of the petroleum and the automotive industries with regard to present and probable future supplies of motor-fuel. There were present at this conference representatives of the two industries, including the National Automobile Chamber of Commerce and the Society of Automotive Engineers on the automotive side, as well as represenatives of the Bureau of Standards and the Bureau of Mines.

After consideration of a number of proposed research projects, the conference decided to concentrate the efforts of the available research laboratories on a single problem that appeared to be of the widest importance to the two industries. Recognizing that the supply of crude petroleum and hence of motor-fuel is limited, and that there is reason to expect the demand to more than keep pace with the supply, it is of importance to us that the price of fuel should be kept at a minimum or the production at a maximum. It is of importance to the petroleum industry that the maximum amount of motor-fuel should be had from the crude-oil supply, since motor-fuel is the highest-priced quantity product of petroleum. The result of the relation between supply and demand has been commercial gasoline, which we have been able to burn with some degree of satisfaction, but with much more satisfaction at some times than at others because its quality has changed.

It has been suspected for a long time and has recently been proved by experiment that as gasoline becomes less volatile, the amount used per mile increases; however, the amount produced per barrel of crude also increases. If we confine our attention to fuel consumption, neglecting for the moment such things as crankcase-oil dilution and the like, it is obvious that there is a balance between the increase in production that can be secured through increasing the "end-point" of gasoline, and the decreased mileage that this fuel will yield; and it may be assumed that the interests of the two industries will be best served when the quality of gasoline marketed is such as to give a maximum mileage per barrel of crude oil used in its production.

It is important to remember also that for the purpose of this discussion we are concerned with the average vehicle in the hands of the average driver, and not at all with what vehicles or drivers might be under ideal conditions, or even very much with what they may be 5 or 10 years hence. It is the average driver of the average vehicle who will consume the next few years' supply of fuel.

To adjust the quality of gasoline so as to meet the condition of maximum utility as defined above, or in fact any other specified condition, it is necessary to know

(1) The relation between fuel consumption and volatility for the average vehicle in use
(2) The relation between volatility and the amount produced per barrel of crude under average refinery conditions

For an answer to the second question we can depend upon information to be secured from the petroleum industry, partly through the agency of the Bureau of Mines. The first question however, is purely an automotive one and for its answer two distinct research projects are in progress.

BUREAU OF STANDARDS TESTS

The first of these, in point of time, is being handled by the Bureau of Standards with the cooperation of the Bureau of Mines. This work was initiated before the organization of the Research Department; hence we, as a department, had nothing to do with its inception. Plans for this research had been under way at the Bureau of Standards for many months and were substantially completed some time ago.

The Department has, however, devoted much time to consultation with engineers and executives in both the automotive and the petroleum industries to insure the necessary financial and moral support for this research and to make the results as useful as possible to the two industries. I do not propose to describe them in detail since W. S. James, of the Bureau of Standards, who is in charge of the work, and other members of the staff of the Bureau are to speak to you about them later. The work has been the result of a year of careful study and experimental development of apparatus and methods. In connection with the work the Bureau of Mines is in charge of the testing of fuel samples, and will compile the results derived from this part of the tests and will assist in the second part of the program as outlined above determining the relation between fuel volatility and the amount produced from the average crude oil.

The principal task of the Society in this connection has been to secure funds for the work of the Bureau of Standards and the Bureau of Mines. This problem was presented to the National Automobile Chamber of Commerce and the American Petroleum Institute as a joint research program, with complete explanations of the nature of the work and the cost of materials, apparatus and assistants. The two organizations voted to share in the expenses, and very hearty promises of cooperation were received from individual companies. A number of research engineers have been secured by loan, one each from different firms representing both industries, and active research work on the program is now in progress.

Supplementary Tests

The second portion of the program was undertaken at the suggestion of a member of the Research Committee to supplement the program of the Bureau of Standards. The latter, consisting of a series of road tests of a limited number of cars under as nearly as possible laboratory conditions as regards precision of measurement, could not include all conditions of use that the average driver encounters. It was thought, therefore, that much might be gained by a series of less elaborate tests which could include a much larger number of vehicles, driven by average drivers in normal service. Accordingly, a supplementary research program was drawn up along these lines, and visits were made to several companies, including those building passenger cars in the largest number, and at present nine different companies are each running, or have completed, a series of tests that will be described later by the engineers in charge of the tests.

The plans included observation of the comparative effect of the four grades of fuel in crankcase-oil dilution, by draining and refilling the crankcases of the test cars each time the fuel was changed. The used-oil samples are to be tested at the Bureau of Standards, as well as by the several companies, to check the amount of dilution that resulted from the use of each of the fuels.

FUELS FOR TEST PURPOSES

The selection of fuels for test purposes presented some rather complicated questions, as to the number of fuels to be used, the range of volatility to be covered, the limitations to be imposed as to chemical composition and as to refining methods. These questions involved numerous conferences and visits to refineries, and resulted in the selection of four experimental fuels ranging in volatility from approximately aviation-grade gasoline to a fuel about as bad as commercial gasoline ever gets.

Provision for supplying these fuels was made by the American Petroleum Institute, the fuels to be sold to the experimental laboratories at a price that will partially distribute the cost of the research between the automotive and the petroleum industries. The supplies of fuel have been made-up by two refineries, one in Chicago, and one on the Atlantic coast.

The tests were intended to show under normal everyday running conditions with average drivers, what effect distinct differences in fuel volatility have on the total fuel used per mile of travel. They serve as a check on the results of the Bureau of Standards' tests, which are run with much greater accuracy than is possible under conditions such as these, but must necessarily include a limited number of cars. In this respect they are similar in plan to the tests described by C. L. Coleman[3] that involve an even larger number of vehicles, but not so many different models. The main feature of the tests was the operation of a number of cars of each of several models, by their regular drivers in the course of their ordinary driving, but each car supplied successively for periods of 1 week with fuels differing by definite steps in their volatility characteristics. An essential point was that the drivers should not know what grade of fuel they were using at any time until the tests had been completed.

It should be noted in connection with these tests that the information desired is not the actual fuel-consumption of the different cars, or different models, but the difference in fuel consumption as produced by the differences in volatility of the four selected fuels. While the seventy-odd drivers of cars of 10 models included in this series might not afford a fair average of the fuel consumption of these cars, it may be expected that they represent much more nearly a fair average of the difference in fuel consumption with the different fuels, and hence afford reliable information as to the relation between average fuel-consumption and volatility for these cars. To make the results applicable to the average fuel-consumption throughout the Country, it will be necessary to average them with respect to the estimated number of cars of each model in use, or perhaps, to be more exact, with the estimated total fuel requirement of each of the several models.

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[3] See p. 145.

The Research Department expects to compile the results of the road tests on this basis as soon as possible after the data are available.

THE DISCUSSION

P. S. TICE:—The avowed purpose of all this research seems to be to arrive at the fuel that will give us the greatest economy. The consensus of opinion appears to be that we have not as yet arrived, in average practice, at a method of carburetion or handling of the fuel in the intake that is capable of giving us the maximum possible utilization; so, are we not really wasting time when we try to judge the merits of several fuels from results obtained with them in carbureting apparatus that is admittedly only indifferently good? It seems to me that the data offered this morning tell more about the carbureting devices used than about the fuels that were presumably on test.

T. J. LÍTLE, JR.:—At the meeting of the American Petroleum Institute at Chicago the automotive industry was told, I understood, that the fuel of the future would be heavier and less volatile than that which we are getting at present, and to get ready for it. I think the most important work to consider in research is getting ready to use the heavier fuels. How many companies have done work along that line? Has Dr. Dickinson information as to how far we may be expected to go in that direction? We were told flatly that it was anticipated that we would have to use fuels of an end-point higher than 500 deg.

fahr., and I am wondering how many companies will be ready to utilize these fuels when they get them.

CHAIRMAN H. M. CRANE:—In my opinion the results of the tests that are now being made will have the greatest possible bearing on the use of still heavier fuels, if we are finally forced to use them. We hope the comparison in actual service conditions of a number of different devices for handling the present fuels, which are fairly heavy, and the results obtained with them, will give us some kind of a curve that will indicate the utilization value of petroleum distillates of different volatility in engines of the present general type; that is, in general, four-cycle engines having float-feed or similar carbureters, various metering devices and more or less simple manifolds, with the application of heat in the quantities available. It is as necessary for us to know that as to know what we would have to do if we had a much heavier fuel. We may make it very plain before we get through that the fuel must be modified to suit the needs of the present general type of engine, or the present general type of engine must be completely changed.

I have spoken many times of the fact that the attempt to use a different type of engine is not a new thing; it is one of the oldest things in the industry. It has been carried out under the urge of an immense financial advantage to be able to use heavier fuel. From the time that we first began using gasoline commercially in automotive vehicles, the spread between the cost of gasoline and the heavier fuels has become greater and greater, at least until very recent years, and it has always been so great as to present a tremendous inducement to any one to use the heavier fuel if he could do it and give the service in so doing. There is the same inducement today. That is the reason I have felt that it is very probable that there is more hope in the proper modification of the fuel to suit the engine that has been developed in service, which is particularly suitable for general service because of its simplicity, than in altering the engine by increasing its complication to a very considerable extent to make it suitable for use with some arbitrary form of fuel.

That is why I am glad to hear that a number of engineers of the oil companies are going to the Bureau of Standards and will meet continually with engineers from the automotive industry, with the result that the knowledge of both ends of the problem will be much more thoroughly disseminated. I know that plenty of us do not half realize the difficulties of the production and marketing of petroleum products. I am equally sure that a great many of the petroleum people do not realize that the present enormous extent of the automotive industry today is based absolutely and entirely on the simplicity of the motive power that is supplied by the present type of engine. It is true that the education which the public is receiving from this simple engine has gradually prepared it to use a somewhat more complicated device, and the public is doing that every day. The present engine for using the present heavier fuel is a far more complicated device than the public had available in 1905 to 1908, and it is using it more or less successfully.

It is necessary to remember also that there are a tremendous number of automobiles in service today, that they are using liquid fuel very rapidly, and that they cannot be changed fundamentally. They can be changed in detail, they can have new manifolds possibly or new carbureters, but they cannot be changed into any other type of engine; they cannot be supplied with any very complicated carbureter-manifold system on account of lack of space under the hood. We all know how difficult it is to apply anything special in the form of a carbureter or manifold to a Ford, because the space is not available to put it in.

If these tests can and do succeed in clarifying the situation regarding the existing cars, the ones that are using 90 per cent of the fuel that is produced every day, they will have performed a tremendously useful service. It is wholly possible that they will result in an overall improvement of only 10 per cent in the use of fuel, but that 10 per cent would be about enough to provide the fuel for 1 year's production of cars and is very much worth getting.

I do not want Mr. Lítle to think that I say we should not do anything along other lines, but also I do not want the feeling to go out that there is imminently possible a sudden change in engine design that will make the use of heavier fuels easy and satisfactory for our kind of work, because I am absolutely certain that there will not be present in any of those engine forms the simplicity that will meet the requirements of the widely distributed use of apparatus that is made by the automotive industry.

PROF. E. P. WARNER:—Is it planned to extend the service tests being made by the Bureau of Standards and the various companies to cover the question of mixed fuels? There is sold regularly a so-called gasoline that contains a large percentage of benzol. In Boston a great many car-owners use it exclusively.

Dr. H. C. Dickinson:—There are as yet no definite plans in this respect. It is rather difficult to lay out a series of tests that will cover a matter in which there are so many variables. Perhaps the most important characteristic of these blended fuels is their anti-knock property, which is a matter that Mr. Midgley has covered. We have considered what could be done in the way of determining the economy of various possible fuels, so far as this can be done in an experimental program, but no definite plans have been made.

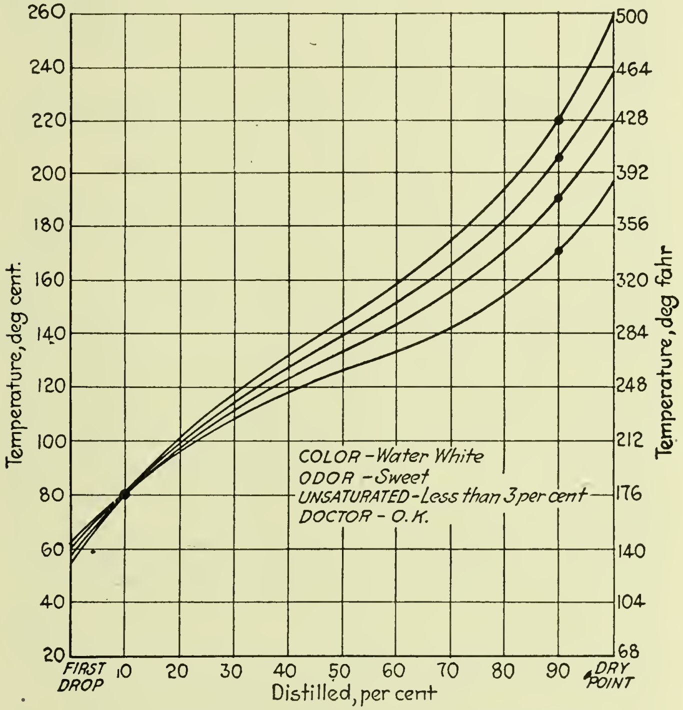

Fig. 1—Distillation Curves of Various Motor-Fuels

I think Mr. Crane has practically answered Mr. Lítle's question, but I would like to add just one thought. You will notice that the upper fuel in Fig. 1 has a 500-deg. end-point. Of course, the fuels were selected before any tests had been made. It was our guess, so to speak, that we had selected the worst fuel which would be likely to "get by" in service without offering so much difficulty that the tests would be meaningless; in other words, we hoped, after securing the results, to plot curves of crude-oil consumption versus fuel volatility which would show a minimum; that we had got beyond the point of economy. Evidently, from the tests, so far as they have gone at present, we were off on our estimates; because the cars that have been tested, and the trucks also, apparently have actually utilized the 500-deg. fahr. end-point fuel without any marked decrease in economy. Of course, that does not cover the question of crankcase-oil dilution, which may be the neck of the bottle; but, so far as the volatility alone is concerned, it looks as if we shall have to make another guess, select two or three heavier fuels and do more testing along the same line before we determine the economic limit of volatility.

W. S. JAMES:—In connection with the point raised by Mr. Tice, there is one advantage in the type of work described by C. T. Coleman[4] that I believe has not been brought out; it is that of fuel specifications. There is considerable controversy at present, at least in the Federal Specifications Board, between the petroleum refiners and the users of gasoline, as to what constitutes a suitable gasoline. The petroleum refiners maintain that the present type of gasoline is satisfactory and that the 90-per cent point can be raised with no detriment to the industry. On the other hand, the users are not sure; they do not know. This was one of the prime reasons the Bureau of Standards took this work up in Philadelphia in cooperation with the Post-Office Department. The Post-Office Department uses something like 4,000,000 gal. of gasoline per year.

There are practically no data on the advantage or disadvantage in actual service of fuels of varying volatility, and these few tests, meager though they are, furnish at least indications as to whether the refiners' demands should be granted. Specifications that were laid down for use in Government purchases have been adopted by a considerable number of municipalities and large companies. This is a matter of great interest to the whole automotive industry. I believe that clarification of the correctness or incorrectness of gasoline specifications is one of the benefits of this kind of work. The Post-Office Department is using Government-specification fuel in about 5500 vehicles in service now. These vehicles represent an investment of capital of about $10,000,000. To change this equipment would be very expensive. It may be easier to change the fuel or the garage operation. Possibly, as Mr. Coleman has indicated, greater gains can be expected from care in carbureter adjustment and in car condition than from change in fuel.

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[4] See p. 145.

P. J. DASEY:—Considering the rapid development of the automobile industry and the number of cars that are being built annually, it seems to me that it will be only a comparatively short time before we shall have such an enormous volume of these vehicles in use that it will be very difficult to secure the proper amount of fuel. To supply the demand as the number of car-users increases, the oil companies probably will have to get a much larger production of crude or give us a larger percentage of the heavier hydrocarbons. If we are to continue producing the same types of engine that we have been building, we will find it increasingly difficult to handle the heavier fuels. It seems that the whole problem resolves into two items. One is that in the development of new engines we base the design on the necessity of using a heavy fuel, up to a 550-deg. fahr. end-point. The other is to continue the work being done by Messrs. Mock and Tice and a number of others and make devices applicable to the present-day types of engine which will enable them to handle the heavier types of engine fuel.

The experiments that are being conducted indicate that we are losing more of the heavier ends in dilution as the fuel becomes heavier. When the fuel is lighter, there is less dilution. On the other hand, we all know that we could get more power from the heavy fuel if we could handle it properly. I believe we will all agree that, if it were possible to handle 550-deg. end-point fuel, the condition would be better because of the much greater volume that would be available from our natural resources.

There is plenty of room for development in engines that would not make them radical departures or increase the cost of production materially, but go a long way toward solving the fuel problem. I will mention only extremely high compressions and operating at lower engine-temperatures. Compression in itself means heat. It has been stated repeatedly that every time the petroleum refiners introduce a series of heavier hydrocarbons into the fuel the engine builders must reduce the compression to handle it. Our experience is just the reverse of that.

I do not know of any oil company that will guarantee to give us now, or at any future time, a 420 or 428-deg. end-point gasoline without raising the price to an enormous figure. The automobile industry will not stand still. The number of cars will increase, and therefore the number of gallons of fuel we shall demand will grow enormously. We cannot grow enormously if we have a high-priced fuel.

Mr. Litle:—I think it would be a mistake to give the oil industry the impression that we can meet them no matter how high the end-point of the fuel goes. Mr. Dasey says he can handle 550-deg. end-point fuel; possibly he can, but we have about 12,000,000 automobiles and trucks running around the Country, and it will never be possible to fix them up. Naturally, that not being possible, they work in a very unsatisfactory sort of way when the end-point goes too high, or when the fuel is not sufficiently volatile.

The oil people absolutely can improve their fuel if they want to. It may cost them more to do it, but it costs the automotive industry much to use the heavy fuel they are proposing to supply. I would rather see them crack more kerosene and mix less kerosene with the fuel. I think it is possible to work along that line, and others to whom I have spoken on the matter agree with that view. I had hoped to hear some one here, representing the fuel industry, say that if we were willing to pay a few cents more for better gasoline we would be guaranteed a supply.

Chairman Crane:—It all comes down to the fundamental thing that the industry is fighting for. About 2 years ago I said that I refused to believe that a straight distillate of crude petroleum was the only fuel that we ought to use in an automobile engine. I did not know much about it, but on general principles I was sure that was to be the case. It took the General Motors Research Corporation to bring clearly before us the fact that it is not necessary for us to do it; that by the addition of various available products, which are not products of petroleum, the fuel that we use in our present engines can be very greatly improved, and a fuel can be made for which an engine of equal simplicity can be designed, wherewith we can expect to get enormous increases in economy and therefore increases in mileage per gallon of crude oil.

I am equally loath to believe that the refining industry has reached a point where its representatives can honestly say that they are giving us the greatest quantity of fuel of a volatility that we can use in present-day engines, that is available in a barrel of crude oil.

I am very glad that Mr. Lítle raised the question of price. I think that has been put too often before the oil people; that we must have the lowest possible price on gasoline, regardless of what the gasoline is. After all, the cost of operating a car is based not only on the cost per gallon, but equally on the mileage per gallon. In other words, the owner really is interested in how much a week it costs him to keep the tank of his car full of gasoline.

The California situation, in which they used distillate and a fairly high quality of gasoline side by side, was very significant. Certain classes of interests that were able to do so applied special equipment to their engines, especially in such cases as motorboat work and heavy trucking, where it could readily be done, and obtained economical operation from the lower-priced distillate, while the other passenger-car owners found they got more satisfaction per dollar expended out of the better grade of gasoline.

I was very much disappointed to hear last year that the refiners had decided to throw both of these products into the same tank and market them as one product. I am absolutely certain that the economic cost to the Country of doing that has been very considerable. They are now supplying a fuel that is not satisfactory for either of the two classes of trade; that is, not as satisfactory as the particular fuel before had been, and the overall price is undoubtedly higher and, equally, the mileage per barrel of crude oil is probably lower. It is the object of all these tests to bring all such facts out into the light.

I fully expect to be shown up as a false prophet by some of the recent tests, as to the ability to handle heavy fuel in some of the more modern cars. It looks as if they were doing it more successfully than I believed they could. I am entirely willing to be shown up that way also. I hope that the oil people, when they find they have been wrong in their contentions, will be willing to say so and show a disposition to change the attitude that they have previously maintained.

T. A. Peck:—I think we are face to face with certain fundamentals that we cannot talk around, desire it though we may. Speaking for the petroleum industry, we can give some of you a 78-deg.-test gasoline, if you want it; but there will probably be 9,000,000 motorists who will not get any, and they are desirous of using their cars. The petroleum industry must have the good-will of the motorist and the automotive engineer. If it is to succeed, it must strive to attain and retain this good will. The petroleum industry today is spending millions in experimenting, in enlarging its refineries, in changing its equipment and in trying to produce enough fuel to keep automotive wheels rolling. My own opinion is that there will not be any radical change in the design of the four-cycle engine we are using today for some time at least. We are all striving earnestly and ably all the time to improve the operation of the engine, because the motorist is becoming more critical every hour. I can remember the day when mixing-valves were good enough for an automobile. We seem to be getting away from the fact that, fundamentally, a carbureter is merely a metering device.

Speaking for the petroleum industry, I hope and believe we all want to do all that we can. For every step that you will take with the engine, we will take two steps to give you the fuel you want. But we must have the raw stock out of which to make it; please remember that it is impossible to get gasoline without crude oil.

DETONATION CHARACTERISTICS OF BLENDED MOTOR-FUELS[1]

By Thomas Midgley[2] Jr and T A Boyd[3]

The effects of admixtures of various percentages of alcohol and alcohol-benzene mixtures for reducing the detonating tendency of paraffin hydrocarbons have been measured by the authors. These results represent an extension of previous work in which similar determinations were made for benzene and other aromatic hydrocarbons. The bouncing-pin apparatus was used for making the determinations. The data obtained by its use are considered to have a high degree of accuracy.

In order that the effects of the blending materials might be measured through as wide a range as practicable, they were blended with kerosene for making the majority of the determinations. This made it possible to ascertain the characteristics of the materials up to a concentration of 80 per cent of benzene or 50 per cent of alcohol without introducing the difficulties due to excessively high engine compression. Because xylidine has the property of exerting a powerful suppressing action on detonation when present in a fuel in percentages that are relatively very small, the standard used as a basis of comparison in the tests was composed of small percentages of xylidine in the paraffin fuel. Tables and curves are appended that show the results of the tests in detail.

That the addition of benzene and other aromatic hydrocarbons to paraffin-base gasolines greatly reduces the tendency of these fuels to detonate when used in automobile engines has been known for some time. Also, it is well known that alcohol when blended with a paraffin-base gasoline improves the combustion characteristics of the fuel. The extension of the by-product coking industry in the United States during recent years has resulted in such an increase in the production of light oil that it can be absorbed only by the use of a part of the material as a motor fuel. This, coupled with the freedom from detonation that characterizes the combustion of benzol in engines, is causing the use of benzol-gasoline blends as motor fuels to be extended as rapidly as the benzol that is available will permit. The scarcity and high price of gasoline in countries where sugar is produced and the abundance of raw material for making alcohol there have resulted in a rather extensive use of alcohol for motor fuel in these districts. As the reserves of petroleum in this Country become more and more depleted the use of benzol, and particularly of alcohol, in commercial motor-fuels will probably become greatly extended.

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[1] Semi-Annual Meeting paper.

[2] M.S.A.E.—Chief of the fuel section, General Motors Research Corporation, Dayton, Ohio.

[3] Assistant chief of the fuel section, General Motors Research Corporation, Dayton, Ohio.

Alcohol as produced commercially dissolves in gasoline only to a very small extent; but the addition of a proper percentage of an aromatic hydrocarbon, such as benzol, toluol or xylol, to the mixture renders the ingredients completely miscible. The availability of benzol for blending with motor fuels makes possible the use of alcohol in such mixtures. Blended motor-fuels containing the three ingredients, alcohol, benzol and gasoline, have been sold in some parts of this Country for some time.

The object of this paper is to report the progress that has been made in measuring the detonating tendencies of mixtures of some of the principal materials that are used as components of the blended motor-fuels now available commercially. The detonation characteristics of aromatic hydrocarbons have been presented in a paper entitled Detonation Characteristics of Blends of Aromatic and Paraffin Hydrocarbons that is soon to be published in the *Journal of Industrial and Engineering Chemistry*. These results have since been extended by determining the detonation characteristics of blends of alcohol and of alcohol-benzol mixtures with paraffin hydrocarbons. Although these data are incomplete and have not been obtained in such a form as to be universally usable, it was thought advisable to present some of them in this way, especially in view of the fact that a considerable amount of the matter dealing with this subject that has been published in the past is in error. As examples of this the following statements by Ricardo may be cited:

(1) Xylene is inferior to toluene for the suppression of detonation[4]

(2) The detonation point of mixtures of ethyl alcohol, acetone, toluene and xylene with paraffin and other

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[4] See *Automobile Engineer* (London) March, 1921, p. 96; also TRANSACTIONS, vol. 17, part 1, p. 21.

> hydrocarbons follows a straight law when the mixture is apportioned by weight and not by volume; that is, the addition of 40 per cent by weight of, say, toluene to hexane would raise the detonation point exactly four times as much as the addition of 10 per cent[5]

The primary reason for the unreliability of some of the data on blending characteristics that have been obtained and published is the previous lack of a means for measuring the detonating tendencies of fuels with sufficient accuracy. The use of the bouncing-pin method for the measurement of the intensity of detonation, however, gives results that are reliable and have a high degree of accuracy. This instrumentation, which is illustrated in Fig. 1 and was described in our paper entitled

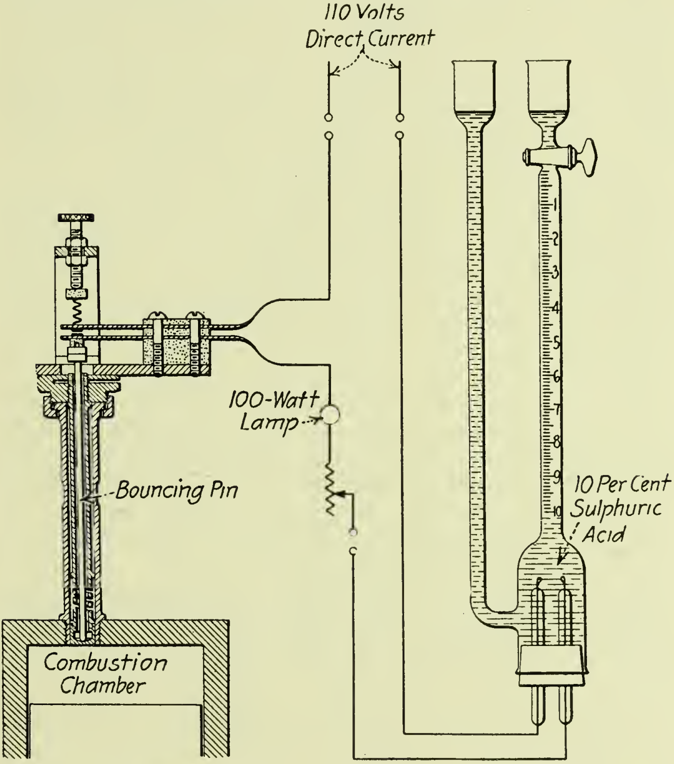
110 Volts
Direct Current
100-Watt
Lamp
Bouncing Pin
Combustion
Chamber
10 Per Cent
Sulphuric
Acid
1
2
3
4
5
6
7
8
9
10

FIG. 1—ARRANGEMENT OF APPARATUS FOR MEASURING DETONATION BY THE BOUNCING-PIN METHOD

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[5] See *Automobile Engineer* (London) March, 1921, p. 94.

Methods of Measuring Detonation in Engines[6] that was presented at the 1922 Annual Meeting of the Society, has made it possible to secure the data that are presented in this paper.

In order that the effects of blending materials might be measured in as wide a range of concentrations as practicable, they were blended with kerosene for making the majority of the determinations reported in this paper. The greater tendency of kerosene than lighter paraffin hydrocarbons to detonate made it possible to determine the detonation characteristics of blends up to a concentration of 80 per cent benzene or 50 per cent alcohol without introducing the difficulties incident to excessively high engine compression. The curves of Fig. 2 show that in general the characteristics of gasoline blends follow closely those of kerosene. This agreement is still better on the molecular basis, as is brought out in the previously mentioned paper on Detonation Characteristics of Blends of Aromatic and Paraffin Hydrocarbons. So that results obtained from a given concentration of blending material in kerosene are applicable within fairly close limits to blends of similar compositions in which the kerosene has been replaced by a gasoline.

On account of variations in engine conditions it is evident that data obtained from any particular engine are applicable in a quantitative way only to that one design and set of conditions. But, although widely different behavior may characterize the combustion of a certain fuel in two different engines, the relative behaviors of two given fuels will be comparative in whatever type of engine they may be run. Hence, in measuring the detonating tendency of any fuel it is essential that some standard be used as a basis of comparison. In the tests reported herein small percentages of xylidine in the same paraffin fuel that was used for blending with the alcohol and with the aromatic hydrocarbons were employed as a standard. Xylidine has the property, common to aromatic amines and considerably more marked in a number of other materials, of exerting a powerful suppressing action on detonation, when present in a fuel in percentages that are relatively very small. Thus, it may be seen from Fig. 2 that 1 per cent of xylidine in kerosene is equivalent for the elimination of detonation to about 15 per cent of benzene in the same material. This property of xylidine makes it possible to convert kerosene into a

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[6] See TRANSACTIONS, vol. 17, part 1, p. 126.

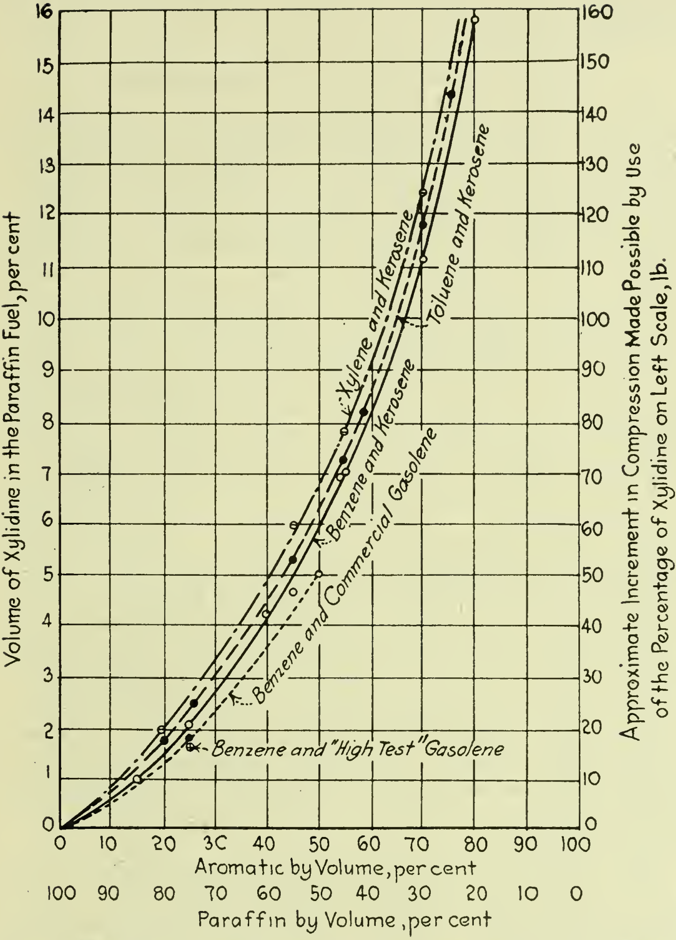

FIG. 2—GRAPHICAL ARRANGEMENT OF THE DATA OBTAINED IN DETERMINING THE DETONATION CHARACTERISTICS OF BLENDS OF AROMATIC AND PARAFFIN HYDROCARBONS

fuel that will withstand very high compressions without knocking, and with the addition of such a small percentage of xylidine that the combustion characteristics of the kerosene, other than its tendency to detonate, are not materially changed.

PROPERTIES OF MATERIALS USED AS FUELS

The materials used as ingredients of the various fuels that were either examined or employed as standards in the examinations, the results of which are reported in this paper and in the previous one referred to above, were "high-test" gasoline, commercial gasoline, kerosene, xylidine, benzene, or 90-deg. benzol, toluene, xylene and alcohol. The xylidine employed was a commercial material composed of the mixed xylidines. The alcohol used was absolute ethyl. The use of absolute rather than commercial denatured alcohol in these tests was necessary on account of the almost complete insolubility of the latter in paraffin oils, unless a "binder" such as benzol is used. Some physical properties of the other materials included in this list are presented in Table 1.

A ¾-kw. Delco-Light engine was used for making all the determinations. This is a single-cylinder, air-cooled engine, direct-connected to a 32-volt, direct-current generator, and having a 2½-in. bore and a 5-in. stroke. The engine was standard, except that a means was provided for adjusting the spark-timing, and that the compression was increased by stages from the normal ratio of 3.47 to 1 to a ratio of 5.36 to 1. This was done by a series of cylinder-heads that had been cut down by different amounts, so as to reduce the clearance volume by corresponding stages. The device employed for measuring the relative intensities of different detonations and called the bouncing-pin apparatus is shown diagrammatically in Fig. 1.

The method used in making the determinations can best be explained by giving a specific example, for which the comparison of a blend containing 45 per cent of benzene and 55 per cent of kerosene with fuels composed of small percentages of xylidine in kerosene was employed. A compression-ratio of 3.87 to 1 was used, so that some detonation would occur, but which was not so violent as to cut down the power of the engine seriously or to cause it to operate in an erratic manner. The fuel under examination was put into one side of the fuel system and the mixing-valve on the engine was adjusted so as to give a maximum of detonation. This adjustment produces almost the leanest possible mixture for maximum power. By trial it was found that 5 per cent of xylidine in kerosene had a slightly less detonating tendency than the benzene-kerosene blend under examination. This fuel was then placed in the other side of the fuel system and its level was adjusted so as to give the point of maximum detonation. The setting of the mixing-valve was left undisturbed throughout the determination so that the compression pressure of the engine would be unchanged. A number of alternate 1-min. runs were then made, with the 5-per cent-xylidine-in-kerosene and the benzene-kerosene blend. The amount of gas

TABLE 1—PHYSICAL DATA ON THE FUELS USED IN THE TESTS

Hydrocarbon	Kerosene	Commercial Gasoline	"High-Test" Gasoline	Benzene, 90-Deg. Benzol	Toluene	Xylene
Specific Gravity at 15 Deg. Cent. (59 Deg. Fahr.)	0.816	0.734	0.704	0.878	0.860[a]	0.860[a]
Absorption in Cold Sulphuric Acid, per cent	7	5	3	....	....	....
Distillation Temperatures						
First Drop,						
deg. cent.	186.0	40.0	44.0	74.0	107.0	135.0
deg. fahr.	366.8	104.0	111.2	165.2	224.6	275.0
10 Per Cent,						
deg. cent.	201.0	65.0	59.0	77.5	108.0	136.0
deg. fahr.	393.8	149.0	138.2	171.5	226.4	276.8
20 Per Cent,						
deg. cent.	207.0	83.5	68.5	78.7	108.5	136.2
deg. fahr.	404.6	182.3	155.3	173.7	227.3	277.2
30 Per Cent,						
deg. cent.	212.0	99.0	76.0	79.2	108.6	136.5
deg. fahr.	413.6	210.2	168.8	174.6	227.5	277.7
40 Per Cent,						
deg. cent.	217.5	111.5	82.7	79.8	108.7	136.7
deg. fahr.	423.5	232.7	180.9	175.6	227.7	278.1
50 Per Cent,						
deg. cent.	222.0	125.0	89.3	80.1	108.8	136.9
deg. fahr.	431.6	257.0	192.7	176.2	227.8	278.4
60 Per Cent,						
deg. cent.	227.5	140.0	96.0	80.5	108.8	137.1
deg. fahr.	441.5	284.0	204.8	176.9	227.8	278.8
70 Per Cent,						
deg. cent.	233.5	157.5	103.0	81.1	108.8	137.3
deg. fahr.	452.3	315.5	217.4	178.0	227.8	279.1
80 Per Cent,						
deg. cent.	241.0	177.0	114.0	82.0	108.9	137.5
deg. fahr.	465.8	350.6	237.2	179.6	228.0	279.5
90 Per Cent,						
deg. cent.	253.5	200.0	128.0	85.0	109.0	137.8
deg. fahr.	488.3	392.0	262.4	185.0	228.2	280.0
95 Per Cent,						
deg. cent.	268.0	219.0	157.0	92.5	109.2	138.1
deg. fahr.	514.4	426.2	314.6	198.5	228.6	280.6
Dry, deg. cent.	291.0	226.0	178.0	....	....	....
deg. fahr.	555.8	438.8	352.4	....	....	....

*Approximate.

evolved in the electrolytic cell during each period was recorded. The output of the generator in volts and amperes was also kept as a matter of record. After three to six runs had been made with each fuel, the benzene-kerosene blend was replaced with 4 per cent of xylidine in kerosene and a second series of runs was made in the same manner. The amounts of gas evolved during the 1-min. runs were then averaged, and the values thus obtained for the xylidine-kerosene fuels were plotted on a coordinate chart having as its vertical axis the amount of gas evolved per minute and as its horizontal axis the percentage of xylidine in kerosene. These two points were next joined by a straight line. From the point at which this line crossed the horizontal line corresponding to the volume of gas evolved by the benzene-kerosene fuel under examination a vertical projection was made to the horizontal scale at the bottom of the chart giving the percentage of xylidine in kerosene. The intersection of this projected line with the bottom scale then gave directly the percentage of xylidine in kerosene that was equivalent in its effect for the suppression of detonation to 45 per cent of benzene in kerosene.

RESULTS

The data obtained in the tests on which this paper is based are given in Tables 2 and 3. The averages of the results given in these tables have been used in plotting the curves on the chart reproduced in Fig. 3. Attention is called to the consistency of the data, and to the agreement between the results obtained with like concentrations of given materials. The close checks that were made in different determinations of the detonation characteristics of a given blend indicate that the values as obtained have a high degree of accuracy.

Fig. 2 gives a graphical presentation of the results obtained in measuring the detonation characteristics of blends of aromatic and paraffin hydrocarbons. These data are tabulated and discussed in the previous paper referred to above. From Fig. 2 the rapidly increasing slope of the curves as the percentage of the aromatic constituent is raised may be noted. Thus the curves show that the presence of only a small percentage of an aromatic hydrocarbon in a paraffin fuel has but a slight effect toward suppressing detonation. This is in agreement with the practical observation made by those who have used benzol-gasoline blends that the addition of less

TABLE 2—DATA OBTAINED IN DETERMINING THE DETONATION CHARACTERISTICS OF VARIOUS BLENDS OF ALCOHOL AND KEROSENE

Determination Number	Compression-Ratio	Spark, Deg. Before Top Dead-Center	Alcohol-Kerosene Blend		Determined Equivalent Xylidine in Kerosene by Volume, per cent	
			Alcohol by Volume, per cent	Kerosene by Volume, per cent	Individual	Average
74	3.47 to 1	43	15	85	2.25	
75	3.47 to 1	43	15	85	2.30	
76	3.47 to 1	43	15	85	2.40	2.30
68	3.87 to 1	32	25	75	4.60	
69	3.87 to 1	32	25	75	4.60	
70	3.87 to 1	32	25	75	4.60	4.60
53	4.59 to 1	32	35	65	7.40	
54	4.59 to 1	32	35	65	7.15	
55	4.59 to 1	32	35	65	7.00	
56	4.59 to 1	32	35	65	7.10	7.20
50	5.36 to 1	25	50	50	12.40	
51	5.36 to 1	25	50	50	12.70	
52	5.36 to 1	25	50	50	12.80	12.60

TABLE 3.—DATA OBTAINED IN DETERMINING THE DETONATION CHARACTERISTICS OF VARIOUS BLENDS OF AN EQUI-MOLECULAR MIXTURE OF ALCOHOL AND BENZENE WITH KEROSENE

Determination Number	Compression-Ratio	Spark, Deg. Before Top Dead-Center	Fuel Blend		Determined Equivalent Xylidine in Kerosene, by Volume, per cent,	
			Equi-molecular Alcohol and Benzene, by Volume, per cent	Kerosene, by Volume, per cent	Individual	Average
77	3.47 to 1	43	20	80	2.35	
78	3.47 to 1	43	20	80	2.25	
79	3.47 to 1	43	20	80	2.15	2.25
71	3.87 to 1	32	35	65	4.65	
72	3.87 to 1	32	35	65	4.75	
73	3.87 to 1	32	35	65	4.75	4.75
80	4.59 to 1	32	50	50	8.60	
81	4.59 to 1	32	50	50	8.20	
82	4.59 to 1	32	50	50	8.60	8.50
84	5.36 to 1	25	65	35	13.15	
86	5.36 to 1	25	65	35	13.55	
87	5.36 to 1	25	65	35	13.45	
88	5.36 to 1	25	65	35	13.30	13.40

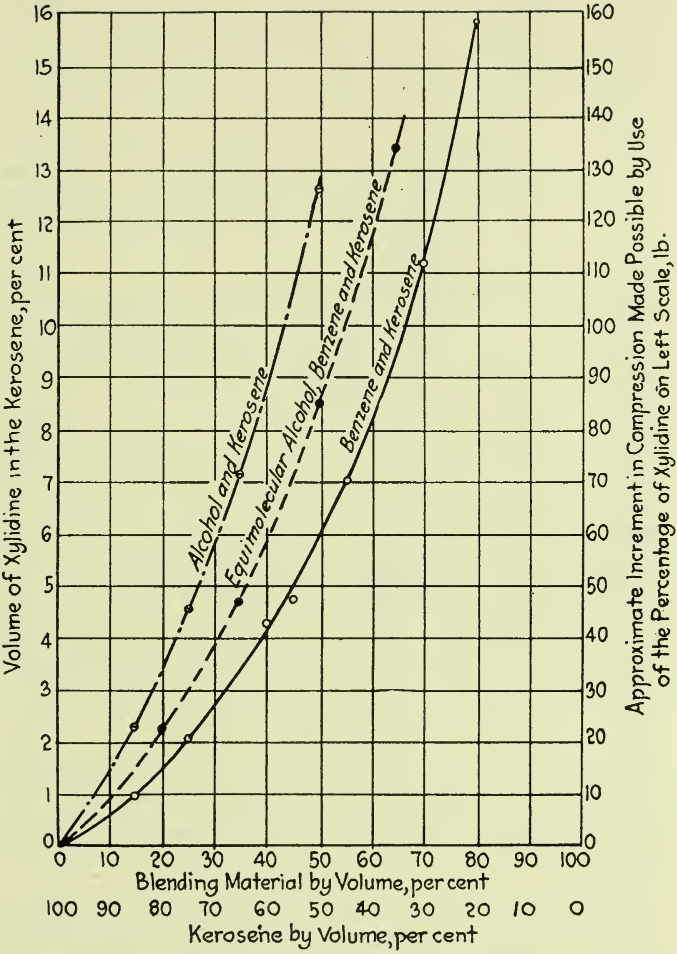

FIG. 3—CHART SHOWING THE EFFECTS ON THE DETONATION CHARACTERISTICS OF KEROSENE OF BLENDING WITH IT VARIOUS PERCENTAGES OF ALCOHOL AND BENZENE

than 20 per cent of benzol to a commercial gasoline or a naphtha exerts only a small influence toward causing the engine to give smoother operation. But when benzol is blended with paraffin fuels in larger percentages its effect increases rapidly as its concentration relative to the paraffin fuel is raised. This is due, in part at least, to the greater percentage of reduction in the amount of the paraffin constituent present as the aromatic content of the blend is increased. It will also be observed from Fig. 2 that toluene on the basis of volume is more effective than benzene for eliminating detonation conditions, and that xylene is, in turn, still more effective than toluene for this purpose.

The vertical scale at the right of Fig. 2 shows approximately the increments in compression pressure of the engine that are made possible by the addition to a paraffin fuel of the corresponding percentages of xylidine given on the vertical scales to the left. From the two scales on the charts it will be observed that the addition of 1 per cent of xylidine to a fuel that gives incipient detonation in a certain engine makes it possible to raise the compression of the engine about 10 lb., without any greater detonation being obtained than with the untreated fuel at the original and lower compression. The increment in compression made possible by each per cent of xylidine added to the fuel can only be approximated, but the values given are based upon a number of observations made under practical operating conditions, on engines ranging from the single-cylinder Delco-Light to the 12-cylinder Liberty, over a compression range of from 50 to 160 lb. By referring the curves given on the charts to the scales at the right, an approximation may be obtained of the relative composition necessary to give smooth operation at a corresponding increase above the normal limiting or critical compression of the paraffin fuel alone.

The data in Tables 2 and 3, together with the benzene-kerosene curve of Fig. 2, are arranged graphically in Fig. 3. This chart shows to a good advantage the relation between the effectiveness of alcohol and that of benzol for the suppression of detonation when blended with a paraffin fuel. It will be observed that on the volume basis alcohol is considerably more effective than benzol for this purpose. Thus, from the chart, 35 per cent of alcohol blended with kerosene produces an effect in suppressing the detonating tendency of the fuel equal

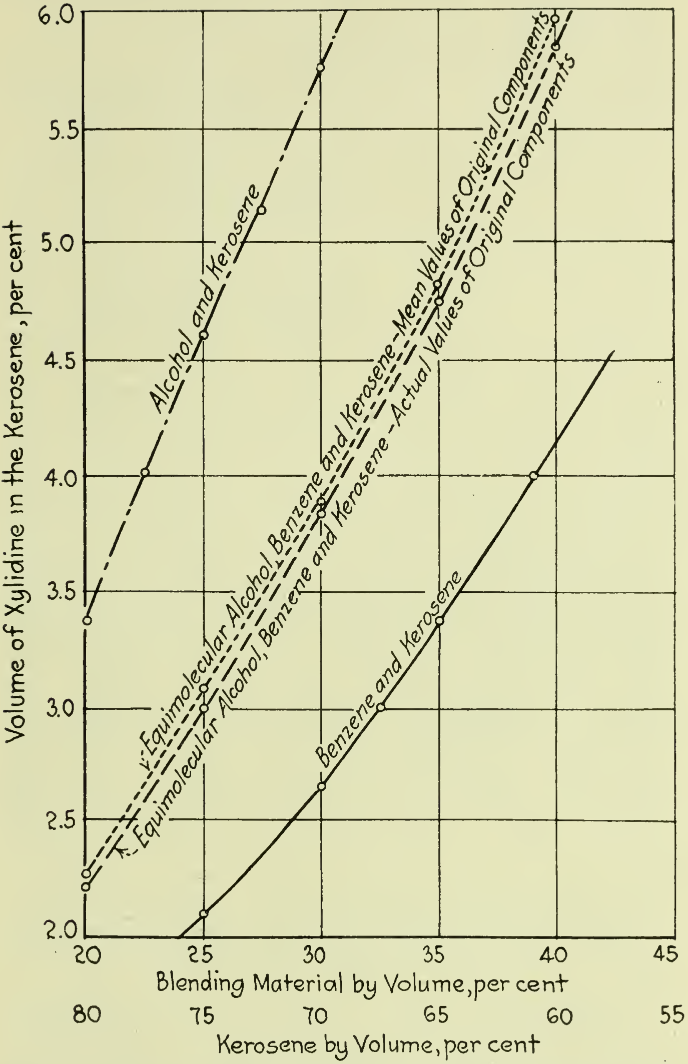

FIG. 4—A BLEND OF TWO FUELS SOMETIMES HAS A GREATER TENDENCY TO DETONATE THAN IS INDICATED BY A MEAN OF THE VALUES OF ITS COMPONENTS

to that given by 55 per cent of benzol blended with the same material.

The middle curve of Fig. 3 is plotted from data obtained in determining the effect of blending an equi-molecular mixture of alcohol and benzene with kerosene in the percentages by volume as indicated in Table 3. This equi-molecular mixture was composed of 39.2 per cent of alcohol and 60.8 per cent of benzene by volume. A given volume of the alcohol-benzene mixture contained, of course, equal amounts of alcohol and benzene on the molecular basis. Thus, 100 cc. of the mixture contained 0.675 gram molecules of each of the ingredients, alcohol and benzene. The curve of the detonation characteristics of the fuel obtained by blending this mixture with kerosene should lie somewhere between the curves obtained in a similar way when using alcohol-kerosene and benzene-kerosene blends, respectively. Since the alcohol-benzene mixture contained 39.2 per cent of alcohol and 60.8 per cent of benzene by volume, it is natural to suppose that for a given concentration of this mixture in kerosene the point representing the detonation value of the blend should lie above the benzene-kerosene curve a distance equal to 0.392 part of the differential between similar points on the benzene-kerosene and the alcohol-kerosene curves. Because they are of such small magnitude no account has been taken here of the changes in volume that occur when some of these materials are blended. In making the mixtures used in the tests, each ingredient was measured separately; that is, before being blended. But, while the actual points lie very close to this mean value, it is significant that in every case they are below it. This statement is illustrated by the curves of Fig. 4 and by the data presented in Table 4. The values given in the first three items of Table 4 were taken directly from the curves of Fig. 3, and those in Item 4 were obtained by multiplying the corresponding values in Item 2 by 0.392. The curves in Fig. 4 are based on those in Fig. 3 and on the figures in Table 4.

It appears, then, that the mixture obtained by blending two fuels of definite detonation characteristics sometimes has a greater detonating tendency than is indicated by the arithmetical mean between the components on the basis of the percentage in which each is present. Attention has previously been called to the fact that in some cases two fuels of similar detonation characteristics, upon being blended, give a fuel that has a very much

TABLE 4—A COMPILATION OF DATA BASED ON FIG. 3 AND ILLUSTRATING THE OBSERVATION THAT A BLEND OF TWO FUELS SOMETIMES HAS A GREATER TENDENCY TO DETONATE THAN IS INDICATED BY THE CHARACTERISTICS OF ITS COMPONENTS

(1) Blending Material in Fuel on the Basis of Volume, per cent	20	30	40	50
(2) Excess Effect of Alcohol over Benzene in Equivalent Percentage of Xylidine	1.900	3.050	4.650	6.650
Excess Effect of Alcohol-Benzene Mixture over Benzene in Equivalent Percentage of Xylidine				
(3) Actual Value	0.700	1.150	1.700	2.550
(4) Mean Value of Components	0.750	1.200	1.820	2.600
(5) Ratio of Value in Item 2 to That in Item 3	0.369	0.377	0.366	0.384

greater tendency to detonate than either of the ingredients.[7] The results obtained in the tests reported in this paper appear to indicate that this characteristic is common to blended fuels; that is, the detonating tendency of a fuel composed of two ingredients is greater than the average of the values representing the detonating tendencies of the two components taken separately. But this is a point that has not yet been determined accurately for a wide range of different materials.

In Tables 5 and 6 and in Fig. 5 are shown the results obtained by converting the percentages of the fuel ingredients by volume to the molecular basis. The compositions in percentages by volume as given in Tables 5

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[7] See Transactions, vol. 17, part 1, p. 135.

TABLE 5—RELATIONS BETWEEN AMOUNTS OF XYLIDINE AND ALCOHOL REQUIRED TO IMPART TO KEROSENE LIKE COMBUSTION CHARACTERISTICS FROM THE STANDPOINT OF DETONATION

Xylidine in Kerosene, by Volume, per cent	In Percentages, by Volume,		In Equivalent Percentages, by Molecules,	
	Alcohol	Kerosene	Alcohol	Kerosene
1.4	10.0	90.0	29.0	71.0
2.3	15.0	85.0	40.0	60.0
4.6	25.0	75.0	55.7	44.3
7.2	35.0	65.0	67.0	37.0
8.8	40.0	60.0	71.5	28.5
12.6	50.0	50.0	79.0	21.0

TABLE 6—RELATIONS BETWEEN AMOUNTS OF XYLIDINE AND EQUI-MOLECULAR MIXTURES OF ALCOHOL AND BENZENE REQUIRED TO IMPART TO KEROSENE LIKE COMBUSTION CHARACTERISTICS FROM THE STANDPOINT OF DETONATION

Xylidine in Kerosene, by Volume, per cent	In Percentages, by Volume,		In Equivalent Percentages, by Molecules,	
	Alcohol-Benzene	Kerosene	Alcohol-Benzene	Kerosene
0.95	10.0	90.0	24.7	75.3
2.25	20.0	80.0	42.5	57.5
3.85	30.0	70.0	56.0	44.0
5.85	40.0	60.0	66.3	33.7
8.50	50.0	50.0	74.7	25.3
13.40	65.0	35.0	84.5	15.5

and 6 were taken from the curves in Fig. 3. In computing the percentage composition of a blend on the molecular basis from its composition by volume the specific gravity and the molecular weight of each of the ingredients were employed. In view of the somewhat wide distillation-range of the benzol used in the tests, the values for which are given in Table 1, a molecular weight of 79 instead of 78 was taken for the benzene. Since kerosene is not a definite compound, and therefore does not have a definite molecular weight, it was necessary to compute an "average molecular weight" for the material. This was done by the method of Wilson and Barnard.[8] For this purpose the distillation data of the fuel given in Table 1 were arranged in the usual type of curve in which the temperature is plotted, on the vertical axis, against the percentage distilled, on the horizontal axis. From this curve the percentages of the fuel distilled in each 10-deg. interval were obtained, and these values were plotted on a chart in which the scale of the vertical axis was in terms of the percentage distilled and that of the horizontal axis was in terms of temperature. The average boiling-point of the fuel was taken as the point at which a perpendicular passed through the center of gravity of the area enclosed under this differential distillation-curve cuts the horizontal or temperature axis. As determined in this way the average boiling-point of the kerosene used in these tests was 226 deg. cent. (439 deg. fahr.). The approximate molecular weight of the kerosene was computed so that it would bear the same proportionate relation to the hydrocarbons next above and below it in the paraffin

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[8] See The Journal, November, 1921, p. 313.

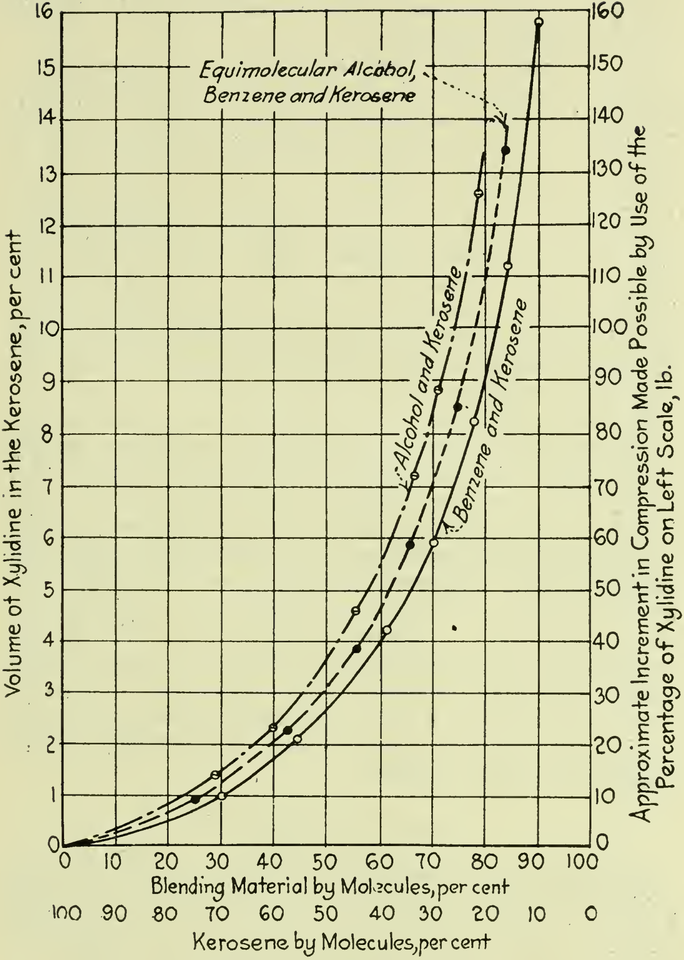

Fig. 5.—Chart Showing the Effects on the Detonation Characteristics of Kerosene of Blending with It Various Molecular Percentages of Alcohol and Benzene

series as the average boiling-point of the fuel bears to the normal paraffin hydrocarbons, dodecane and tridecane, which occupied like positions with respect to it. Obtained in this way, which is recognized as giving only a close approximation, the average molecular weight of the kerosene was 178.5.

On the molecular basis there is not so marked a difference between the effectiveness of alcohol and that of benzol in suppressing detonation as shown in Fig. 5 as there is on the volume basis as indicated in Fig. 3. The closer agreement between the effects of the two materials on the basis of molecular concentration is due to the smaller size of the alcohol molecule as compared with that of the benzene molecule. But even on this basis alcohol is still more effective than benzene for suppressing detonation. Thirty-five per cent of alcohol, which, as is indicated above, is equivalent in effect to 55 per cent of benzene on the volume basis, is equivalent to 42 per cent of benzene on the molecular basis.

Since the middle curve in Fig. 5 is based on the results obtained by blending an equi-molecular mixture of alcohol and benzene with kerosene, it is natural to suppose that any point on it should lie half-way between the points occupying like positions on the two outside curves, which were obtained by blending alcohol and benzene separately with kerosene. However, the points on the middle curve do not occupy this middle position; but, as is the case on the volume basis, as shown in Table 4 and Fig. 4, they are uniformly lower than the mean values of the original components, thus showing that in this case a blend of the two ingredients is not so effective for the suppression of detonation as the mean average of the effects of the ingredients would indicate.

THE DISCUSSION

PRESIDENT B. B. BACHMAN:—It seems to me that Mr. Midgley has shown that the dilution of kerosene with benzene is more effective than the dilution of high-test gasoline with benzene. Is that correct? If so, why?

THOMAS MIDGLEY, JR.:—Yes, the curve as plotted shows the increase of compression pressure that the fuel would stand. The benzol originally would stand a compression pressure of 50 lb. per sq. in. and the commercial benzene 20 lb. per sq. in. For sake of argument, let us say that the kerosene will now stand 70 lb. per sq. in. The increment is not so great and, as the fuels get better, the increment apparently decreases.

P. J. DASEY:—Do the increments of compression shown on the right-hand side of Fig. 2 in the paper indicate an addition to the normal compression, on the basis of calculation? What is that basis of calculation?

MR. MIDGLEY:—In this particular engine the compression pressure is 52 lb. per sq. in.; in the case of gasoline it is 75 lb. per sq. in., or thereabout. It is the increment to whatever the fuel stands.

MR. DASEY:—I noticed that they all started from a common point.

MR. MIDGLEY:—Yes.

MR. DASEY:—It is interesting to have that basic compression pressure put in.

MR. MIDGLEY:—It is difficult to get a reading that is really accurate or to state the compression pressure that the fuel will stand, or the pressure ratio, for the reason that when the engine starts you get a certain value; tomorrow you get a different value; and the next day still another value. You can get any value for any set of barometer readings, carbon deposits or temperature readings. It is affected by the smallest thing. You cannot take a reading and say, "That is it." The increments depend upon the relation between the different fuels, one being much better than another. That is the difficulty of getting down to an accurate basis.

HERBERT CHASE:—What method did Mr. Midgley follow in setting the spark when measuring detonation? I ask this because detonation is known to vary considerably with the spark position.

MR. MIDGLEY:—We set the spark, roughly, at the point of maximum power. On the other hand, when comparing two fuels, the spark-advance was the same for both; so that the effect is cancelled. If the spark were wrong for one, it would be wrong for the other; the method is essentially a comparison of two fuels, one being a standard of some sort. The actual readings show the relation between two fuels; the spark-advance is kept precisely the same in making the comparison.

THE HOT-SPOT METHOD OF HEAVY-FUEL PREPARATION[1]

By F C Mock[2] and M E Chandler[3]

The development of intake-manifolds in the past has been confined mainly to modifications of constructional details. Believing that the increased use of automotive equipment will lead to a demand for fuel that will result in the higher cost and lower quality of the fuel, and being convinced that the sole requirement of satisfactory operation with kerosene and mixtures of the heavier oils with alcohol and benzol is the proper preparation of the fuel in the manifold, the authors have investigated the various methods of heat application in the endeavor to produce the minimum temperature necessary for a dry mixture.

Finding that this minimum temperature varied with the method of application of the heat, an analysis was made of the available methods on a functional rather than a structural basis. Three of these are discussed: (*a*) When the heat from the walls of the manifold is applied through the medium of the air; (*b*) when it is applied to the fuel alone, or partly to the fuel and partly to the air; and (*c*) when a spray of atomized fuel and air is directed against a heated surface. A device was constructed by which the three main variables, the exhaust temperature, the exhaust flow and the area of the heating surface, might be regulated and the three remaining variables, the quantity of air, the quantity of fuel supplied and the quantity of fuel vaporized, might be controlled.

Taking into account the wide range of temperatures that the air charge and fuel supply undergo before entering the intake-manifold system, a quantitative computation of heat transfer was made and the conclusions were drawn that only by a combination of centrifugal force, surface tension and the force of gravity could the unvaporized drops be separated from the fuel charge and that the conditions of combustion are governed by the rate of fuel feed from the manifold to the cylinder and not from the carbureter to the manifold.

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[1] Semi-Annual Meeting paper.

[2] M.S.A.E.—Research engineer, Stromberg Motor Devices Co., Chicago.

[3] M.S.A.E.—Engineer of carbureter design and development, Stromberg Motor Devices Co., Chicago.

Recent years have witnessed increasing attention to the design of intake-manifolds and to the varied methods of handling automotive fuels in preparing them for introduction into the combustion-chamber. The resulting development, however, has been limited in direction, being confined usually to slight modifications of the construction that has been followed ever since automotive engines began to have more than one cylinder. The improvement in economic conditions that all authorities agree is approaching will certainly result in considerably increased use of automotive equipment, and it is not impossible that the demand for motor fuel may bring about a condition of higher cost and lower quality. As it might require three or four years to develop a change of design to meet such a change of fuel, it would seem that now is a fitting time to make a survey of the problems involved in the preparation of our present fuels and of heavier ones and to make a fresh analysis of the situation, entirely apart from and unhampered by the conditions of previous practice.

It is true that many 1922-model cars have operated satisfactorily with the motor fuels at present in use, both in summer and in winter, but many have not. We are convinced that it is possible to operate on mixtures of gasoline, kerosene and some heavier oils, combined with alcohol, benzol or other anti-knock component, as well or better than a number of cars today operate on gasoline, by the use of improved methods of fuel preparation in the intake-manifold.

SPECIFIC REQUIREMENTS OF FUEL PREPARATION

The requirements of proper fuel preparation are

(1) A thoroughly and continuously homogeneous mixture of fuel and air with no drops or liquid-film wall-flow to the valve ports
(2) The charge temperature should be the minimum possible while complying with requirement (1)
(3) The provision for a prompt change in the rate of action under changes of load and speed

A cylinder charge of fuel is only a medium-sized drop. Any one who has observed, through glass manifold sections, the storm of drops that is usually present, can easily appreciate the importance of this point. All the oil dilution in the crankcase is due, of course, to the introduction into the combustion-chamber of fuel that is not burned later. We believe that a large part of the rapid carbon formation, characteristic of engines having poor distribution, is due to the cracking, without burning, of the drops of excess fuel that occasionally enter the cylinders. The first requirement would include the prevention of liquid gasoline from reaching the cylinders after the use of a primer or choke means of starting. As starting is really an increase of the load from zero, devices for starting and quickly warming-up come under the last requirement.

METHODS OF HEAT APPLICATION

If we consider as our objective a minimum temperature of the dry mixture, that is, a mixture of transparent fuel-vapor and air, it is immaterial, in theory, whether the heat is applied first to the fuel or to the air. If, however, we accept what our experiments have apparently demonstrated, that is, that a fog mixture of condensed vapor and air is satisfactory, provided the cylinder temperatures are such as to change this fog to a vapor before the end of the compression stroke, we shall find, both in theory and in practice, that the minimum temperature that can be used will vary with different methods of heat application. The theoretical considerations involved are, we hope, clearly shown by an analysis of the known and available methods of heat application. These have been classified as follows:

Case No. 1. Heat imparted to the mixture through the medium of the air, by the communication of heat from the manifold walls to the air and to such part of the fuel as has been deposited on the manifold walls. This is considered to involve only the production of a dry-vapor mixture. An interesting variation of this is shown as Case No. 1a, where part of the preheating of the air is accomplished by subtracting heat from the air and the vapor mixture already formed, thus giving a fog mixture

Case No. 2. Application of heat first to the fuel alone, with resulting condensation of the vapor when it joins the main air-column; this results in a fog mixture

Case No. 2a. Heating the fuel and a part of the air to generate a rich dry-vapor mixture, which is then condensed as it enters the stream of the remaining air-supply. This gives a fog mixture

Case No. 3. Directing a spray of atomized fuel and air against a heated surface. One result obtained with this construction is the breaking-up of the spray drops into even smaller drops in the so-called "spheroidal" condition; the mixture thus formed can scarcely be properly designated as a fog mixture

This classification has been made on a functional rather than a structural basis. Most of the hot-spot constructions in actual use employ two, and sometimes three, of these heating methods, but for analysis the distinction we have made seemed necessary. Consideration of the direct application of heat to the fuel has been purposely limited to designs in which the fuel has been previously metered in a liquid state, as doing so after heating has not thus far been demonstrated as practicable.

The computation of the mixture-temperatures is based upon the methods used and determinations made by Professor R. E. Wilson and described in THE JOURNAL.[4] The gasoline values used are those of the high end-point gasoline referred to in Professor Wilson's discussion at the 1921 Semi-Annual Meeting.[5] This gasoline, by the way, is apparently quite similar to the *D* gasoline of the fuel research consumption test recently concluded.

HEAT APPLIED DIRECTLY TO THE AIR

In Case No. 1 the air charge receives a heat supply such that after the latent heat of evaporation has been supplied to the fuel, the resulting mixture will have the minimum temperature of a dry vapor. A typical construction is shown in Fig. 1. Its practical equivalent, the application of heat to the air charge before it enters the carbureter, is shown in Fig. 2. With the complete evaporation of a 15-to-1 mixture of the high end-point gasoline measured by Professor Wilson, and ignoring the heating of a small amount of fuel on the walls, this would involve air entering the carbureter at 181 deg. fahr. with a resulting mixture-temperature of 135 deg. fahr. For the kerosene measured by Professor Wilson, the air would have to enter the carbureter at 283 deg. fahr. with a final mixture-temperature of 230 deg. fahr.

In practice, however, dry mixtures are not realized at such low temperatures, for the reason that only part of the hot air comes into contact with the fuel. Within a short distance from the carbureter jet the tiny droplets of fuel spray take up a velocity and direction identical with that of the air which bears them and thenceforth, until they strike a wall, they generally are surrounded by a miniature atmosphere of vapor at the dew-point. Fuel that travels along the walls comes into actual contact with only a thin film of air. We have endeavored by various means to create a turbulence that would accelerate and decelerate the spray droplets in the air medium that carries them, but every effort of this kind has resulted in increased deposition of fuel on the manifold wall and has made conditions worse than before. The temperatures actually existing in practice are more nearly those that would result if the fuel came into heat-conducting contact with but one-half to one-third the air charge during the travel through the intake-manifold. On such a basis the average temperature of the mixture is considerably higher, for instance, with a 15-to-1 dry mixture, and, if the fuel receives heat from one-half the air, the final average temperature will approximate 175 deg. fahr. with

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[4] See THE JOURNAL, November 1921, p. 313, and January 1922, p. 65.

[5] See TRANSACTIONS, vol. 16, part 2, p. 257.

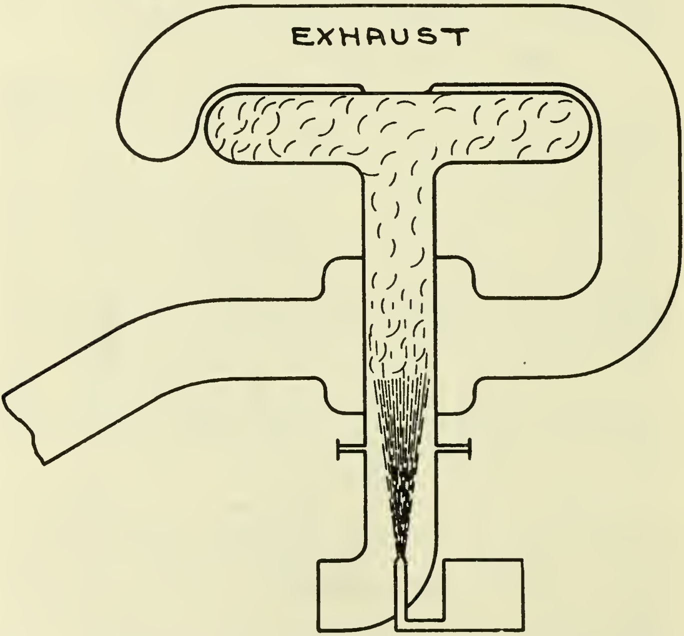

Fig. 1—In This Construction the Heat Is Applied Directly to the Air Portion of the Cylinder Charge

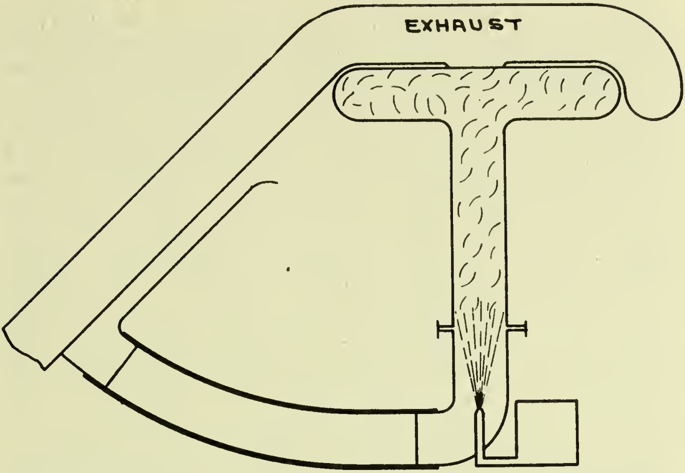

FIG. 2—MANIFOLD CONSTRUCTION TO SUPPLY HEAT TO THE AIR CHARGE BEFORE IT ENTERS THE CARBURETER

gasoline and 276 deg. fahr. with kerosene, as is brought out in Table 1.

TABLE 1 — FINAL AVERAGE MIXTURE-TEMPERATURE WITH ENTERING AIR AND FUEL AT 75 DEG. FAHR.

Fuel	High End-Point Gasoline		Kerosene	
Air-Fuel Ratio	12 to 1	15 to 1	12 to 1	15 to 1
When Heat Is Transferred from All the Air to the Fuel (Fig. 2)	145	135	240	230
When Heat Is Transferred from One-Half the Air to the Fuel (Fig. 2)	195	175	299	276
When Heat Is Transferred from One-Half the Air to the Fuel (Fig. 3) Followed by Cooling of the Charge by the Intake Air	140	128	196	176
When Heat Is Applied Directly to the Fuel (Fig. 4)	143	132	174	159
When the Fuel Only Is Heated to Temperature of Required Vapor Density (Fig. 6)	98	92	124	116

On account of the high heat-capacity of dry-mixture charges formed in this way, there being no cooling from any further evaporation of the fuel during the compression-stroke, the tendency toward detonation should be, and apparently is, greater with this method of fuel preparation than with most others. Due to the relatively slow heat-transfer, more than the customary difficulty is experienced during changes of engine speed and load. The proper functioning of a device of this kind is contingent upon the maintenance of adequate temperatures; but in actual practice such temperature regulation is disturbed by a number of factors, depending upon seasonal and climatic conditions, as will be explained later. Since the mixture-temperature depends upon that of the air entering the carbureter, which in most cars depends in turn upon the temperature of the cooling water and of the whole mass of metal under the hood, there is a long duration of "warming-up" which can be taken care of only by elaborate thermostatic devices. A factor of safety, to provide for the occasional use of fuels heavier than the average, can be obtained only by raising still farther the temperature of the fuel charge of normal operation. More important is the fact that there is nothing to prevent raw gasoline entering the cylinders during the starting and warming-up period and probably also during normal running.

In Case No 1a, Fig. 3, the fuel vapor is formed as in Case No. 1, but a smaller exhaust air-heater is used. The air entering the intake system, before it reaches the exhaust heater, is used to cool and condense to a fog the dry mixture coming from the carbureter. The temperatures of the air entering the carbureter and of the mixture leaving the carbureter are the same as in Case No. 1, but the final mixture-temperature in the intake-manifold, if a complete heat-transfer could be established, would be considerably lower than in Case No. 1; for instance, 128 deg. fahr. with gasoline as against 175, and 176 deg. fahr. with kerosene as against 276. But we do not believe that in practice the addition of this condensing device would be of value. If made elaborately enough to accomplish the desired heat-transfer, it would probably increase the amount of fuel on the walls and require a still higher temperature of the air entering the carbureter. It would also increase the difficulties of acceleration and the "loading" in the intake-manifold while the engine is cold.

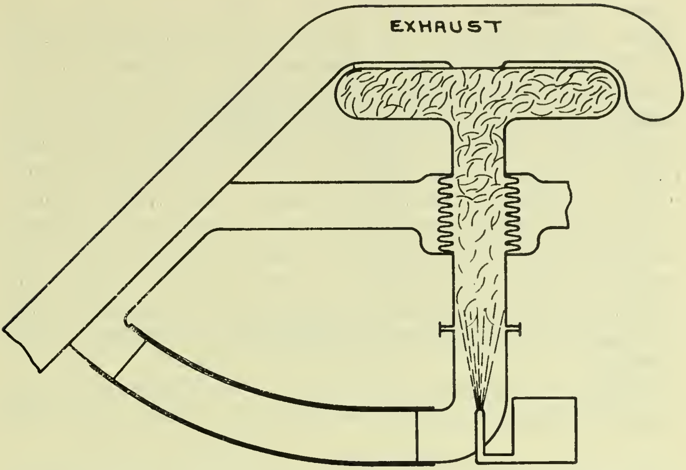

Fig. 3—In This Construction Part of the Initial Heat Application Is Obtained by Cooling the Vapor Mixture to a Fog

Heat Applied Directly to the Fuel

In this method, which is shown in Fig. 4, the fuel, after being metered is discharged into a heating chamber that the air charge does not enter; the vapor formed here is then mixed with the unheated air-charge to form a true fog-mixture. At first thought this system seems to be promising, but actually it has serious inherent disadvantages, for the reason that the delivery of vapor depends upon the temperature being kept above a certain minimum.

A homely illustration of the difficulty of evaporation with this type of heater is afforded by the example of a covered kettle or pot of water maintained at a temperature slightly below the boiling point, say 208 deg. fahr. As any housewife knows, a kettle can be heated in this manner for a long time without losing much water, the reason being that, although the evaporation from the surface of the water is rapid for a while, until the space above the water and beneath the lid becomes filled with vapor, there is no difference in pressure between the vapor and the outside air and no marked escape of water vapor from the spout. It is only when the temperature is raised to the boiling-point that the vapor pressure is able to rise above that of the atmosphere and create a continuous outflow of steam. An open chamber will evaporate liquid below the boiling-point much more quickly than a closed one, the difference being due solely to the more rapid escape of the vapor from the open chamber. In the design illustrated a normal flow of vapor from the heating chamber should take place only after the vapor temperature has been raised to the final boiling-point of the fuel; the vapor must be between 400 and 500 deg. fahr., which is much higher than the temperature needed with any other construction shown. The final temperature of the mixture may, however, be rather low because of the fact that very little more heat need be added to the system than is necessary to vaporize the fuel. Also, the "factor of safety" in heating capacity may be large without raising the final temperature of the mixture in proportion.

This arrangement might be hard to start and would possibly be slow on acceleration. With heavy fuels there

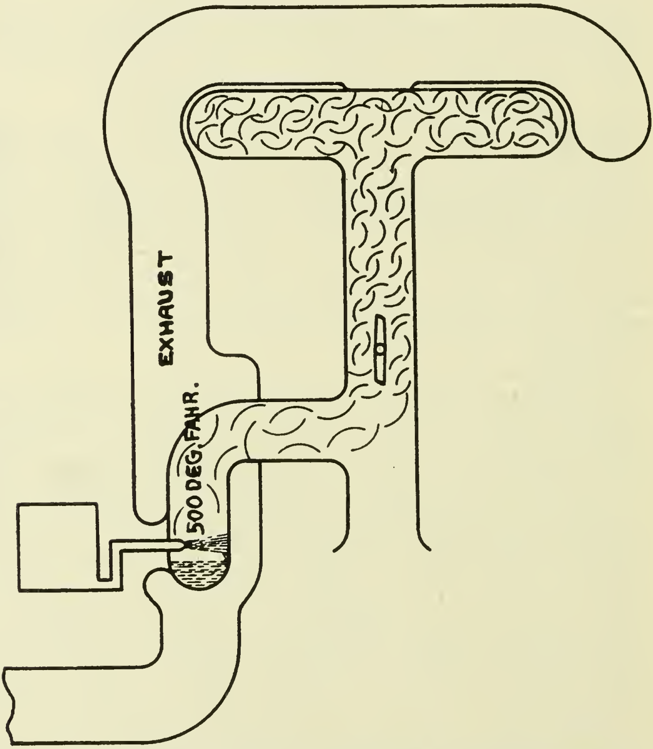

Fig. 4—In This Construction the Heat Is Applied Directly To the Fuel Portion of the Mixture Charge

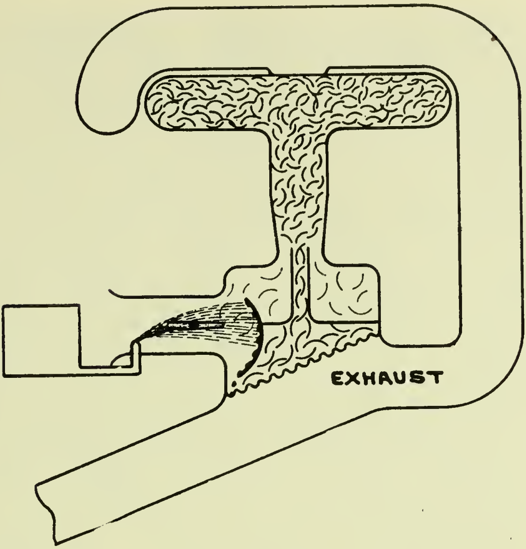

Fig. 5—In This Manifold Arrangement the Heat Is Applied Directly to the Main Portion of the Fuel and a Small Portion of the Air

would be a tendency for the heavy elements to collect in the bottom of the heating chamber during idling, when the exhaust temperature is lower than the boiling-point of the fuel. Upon a sudden increase of the exhaust temperature this pool of heavy elements is apt to coke. In fact, we have known of a number of instances where a pocket for the collecting and heating of the fuel would fill with greasy tar or coke. This trouble was particularly marked with high end-point gasolines.

This construction has the additional advantage, when properly designed, of permitting no liquid fuel to reach the valve ports; on this account, as also with Case No. 2a, it will give a homogeneous fuel charge, or "good distribution," as we call it, with any shape of intake-manifold and any convenient location of the carbureter.

Case No. 2a, Fig. 5, is a sort of compromise between Cases Nos. 1, 2 and 3, which seems to possess all the advantages of Case No. 3 and fewer disadvantages. The mixture spray from the carbureter is thrown against a deflecting surface, which may be heated, and the fuel not vaporized is thrown down into a heating chamber as in Case No. 2. An opportunity is afforded for the fuel to evaporate in and mingle with the air, before the separation of the liquid and the vaporized portion. This reduces the fuel lag on acceleration and also reduces the amount of fuel that must be taken into the heating chamber. An air circulation is maintained through the heating chamber, which helps to carry the vapor away as fast as it is formed; the action in the heating chamber then can be *evaporation* rather than *boiling* as in Case No. 2. This distinction is important because *boiling* implies the maintenance of temperature above a certain point, at all engine speeds and at a constant pressure, while evaporation can take place at any temperature and, fortunately, under a change of engine speed, the decrease of the exhaust temperature is accompanied by a reduction of the fuel feed and the rate of evaporation required.

The air taken through the heating chamber is, of course, highly heated, so that, as compared with Case No. 2, we have a small part of the fuel and of the air heated, to be cooled by the remainder of the air charge and a certain part of the fuel charge. The temperature balance would, of course, depend on the percentages of the fuel initially vaporized and of the air passed through the heating chamber.

This arrangement possesses the advantage of Case No. 2, in allowing a large reserve capacity for warming-up without excessive heating of the mixture under normal operation, and also of preventing liquid fuel from going into the engine cylinders. A device of this sort, though of design entirely different from Fig. 5, has been used in the actual driving of a passenger car with a six-cylinder engine and gave as good a demonstration on kerosene, with a benzol component to avoid detonation, also alcohol at a mixture-temperature of 120 to 140 deg. fahr., as with gasoline. It was also found possible to use heavier fuel combinations, which resulted in perhaps better operation than that shown by the average car in the hands of its owner. One of these mixtures was one-third benzol, one-half kerosene and one-sixth Mobiloil B lubricating oil; another one-fifth alcohol and four-fifths 38 to 40-deg. Baumé distillate, a light oil that cannot be ignited by itself with a match in the atmosphere at the ordinary temperatures and will burn slowly from a wick with a very smoky flame. With these latter mixtures the mileage per pound was not so good as with gasoline or kerosene, and there was a perceptible carbon-deposit; also a slight slowness, but not hesitation, on acceleration. The operation of the car in general was so good that it would easily satisfy the average car-owner, were it not for the necessity of starting on a different and lighter fuel. Starting on gasoline in very cold weather was not more difficult than with the ordinary carbureter and intake-manifold arrangement. In fact, no difficulty was ever experienced in starting; the starter was always strong enough to turn the engine over, and closing the choke would always effect a start. On gasoline the warming-up was very good. In weather 10 deg. fahr. above zero, it was necessary only to use the dash mixture-control device for about ½ min. or less after starting, after which it was possible to set all the controls in the normal driving position and drive away. This usually synchronized with the development of a mixture-temperature of about 90 deg. fahr. With gasoline the fuel-consumption was but slightly lower on a gallon test than with a good carbureter on a conventional type of hot-spot intake-manifold, but the engine would run smoothly on very lean mixtures and the weekly mileage, particularly in winter, was better. The smoothness and the absence of carbon, crankcase-oil dilution and ignition trouble were marked. We found also improved operation at low speed on hills. The engine would pull smoothly and without apparent effort and maintain this smooth low-speed pulling indefinitely.

AIR AND FUEL CHARGE PROJECTED AGAINST THE HOT SURFACE

As illustrated in Fig. 6, this includes a condition aimed at, and more or less realized, in many hot-spots in use today. It is the general belief, perhaps, that the fuel spray strikes the heated surface, vaporizes, and then condenses in the airstream. More recent observations lead us to believe that not all of the fuel vaporizes on the heated surface. It seems that under some conditions the sudden application of heat to one side of the drops of spray, as they strike the heated surface, relieves the surface tension that holds them in globular form and causes them to burst; meanwhile, if an air-draft is present, the "spheroidal condition" keeps them from adhering to the heated surface. This belief was first suggested by the observation that large drops come off such a hot-spot in a coarse spray, while small drops come off in a finer spray.

There is one interesting hypothesis of action under these conditions, the realization of which would give a fog mixture at very low temperatures with a very simple structure. If the heating surface were of exactly the size and location to be wholly covered by the liquid of the fuel spray; if its heating capacity were such that it could vaporize all the fuel that strikes it; and if the scouring action of the air-draft across the heating surface were sufficient to carry away the vapor as fast as it was formed, it would be possible to produce the vapor at the relatively low temperature corresponding to a density of one-fifteenth to one-twelfth that of air; also, there should be little, if any, heat transmitted directly to the air from the heating surface. Under such conditions, which we believe can be realized only in theory, the mixture temperatures would be the minimum among all the systems suggested for producing a fog mixture by external application of heat energy.

Fig. 7 is an effort to show the nature of such action,

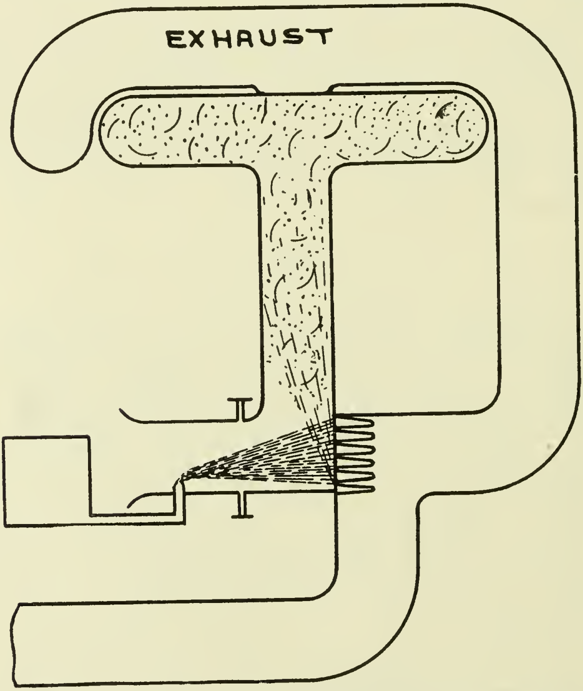

Fig. 6—In This Manifold the Air and Fuel Charge Is Projected Against a Heated Surface

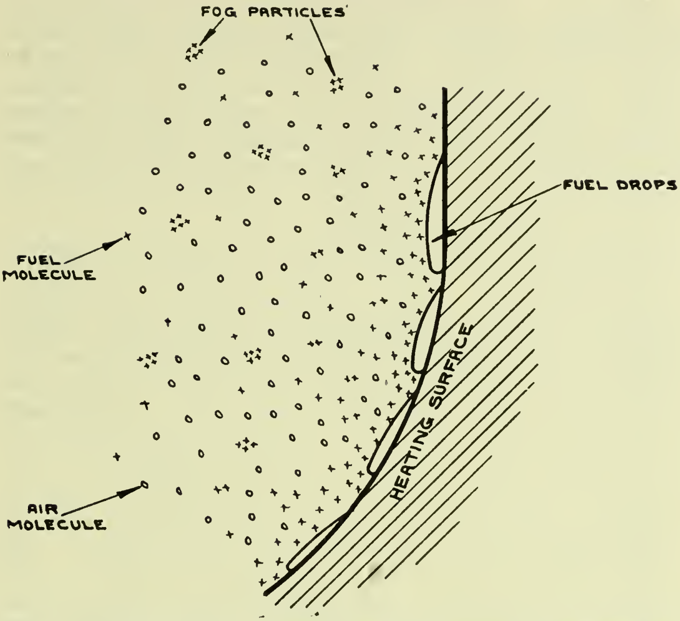

Fig. 7—Diagram Showing the Nature of the Action of the Construction Illustrated in Fig. 6

assuming complete evaporation at the surface. There is, first, near the surface, a film of liquid, or a layer of liquid drops. Just off the liquid film the greatest vapor-density occurs, but as the distance increases and the air begins to lower the temperature, the molecules will begin to gather in droplets, in the action that we term condensation. It is obvious that it would be impossible to bring *all the air* into such contact with the liquid film that the vapor would be swept away, and uniformly diffused, within a few molecule paths of the liquid film; and it is only under such a condition that the temperature balances of Fig. 6 could be obtained. But it also is clear that the more completely we can direct and diffuse the air charge on the heating surface in the conventional hot-spot design, the lower the temperature and density can be next the liquid film, the lower can be the temperature of the liquid film and the wall itself and the lower the final temperature of the charge.

Regardless of the correctness of the theory of operation of this type of hot-spot, there are several advantages and disadvantages in practice that should be pointed out. As already outlined, reserve capacity can be obtained only by making the surface larger. Also, there is no inherent characteristic of this arrangement that would prevent liquid fuel from going into the engine. The heat capacity of the wall of any structure that could be used would be sufficient to prevent any lag in acceleration, provided the carbureter were made to give a charge of slightly increased richness with a fuel of graduated volatility.

Experimental Determination of Heating Action

The foregoing analysis indicated the great importance of several considerations not previously investigated in the problem of properly preparing the fuel. We undertook, therefore, to build an experimental device that would allow us to regulate the three main variables governing the heat input: (*a*) the exhaust temperature, (*b*) the exhaust flow and (*c*) the area of heating surface; and to control the remaining variables affecting heat absorption: (*d*) the quantity of air (*e*) the quantity of fuel, and, so far as possible, (*f*) the vapor density. The device used is shown diagrammatically in Fig. 8. The heating element was an iron plate, 3½ in. wide and about 8 in. long, exposed to the exhaust on the ribbed lower side, and receiving fuel from a series of jets placed across the width of the plate at the air-intake end of its top surface. A slab of magnesia cement encased in a thin metal cover was used as a movable heat-insulating shield to give any desired area of heating surface up to the maximum. Thick layers of asbestos gasket were used to insulate the air-chamber from heat contact with the heating-plate. The fuel flow took place under gravity head and was regulated by hand at each speed to give the desired mixture-strength, the air supply being metered by the orifice-plate method. The fuel-jets could be set to give either a fine or a coarse spray and could be made to discharge in any desired direction. One side and both ends of the device were fitted with glass windows so that the liquid and fog conditions existing within could be easily observed.

It will be noted that by closing air entrance *a* and directing the fuel spray along the length of the plate, the conditions involved in Case No. 3 of our analysis could be reproduced. By using air entrance *a*, the conditions of Case No. 2 or No. 2a would be given, according to whether air entrance *b* was entirely or partly closed.

And if the fuel spray were directed along the passage, instead of at the hot plate, with the air supply taken through entrance *b*, the conditions of Case No. 1 would exist.

The first test-runs with this device indicated that a broader range of investigation than initially planned was advisable. It was mounted in the center of the exhaust manifold of a Continental Model 7-R 3¼ x 4½-in. six-cylinder engine, which was operated for several weeks with this device, on motor gasoline, high-test or aviation gasoline, kerosene and grain alcohol. It should be stated that this form of heater, having a horizontal heating surface, was selected not becauase it was a desirable form for regular use in automotive service, but because of the convenience with which observations could be made.

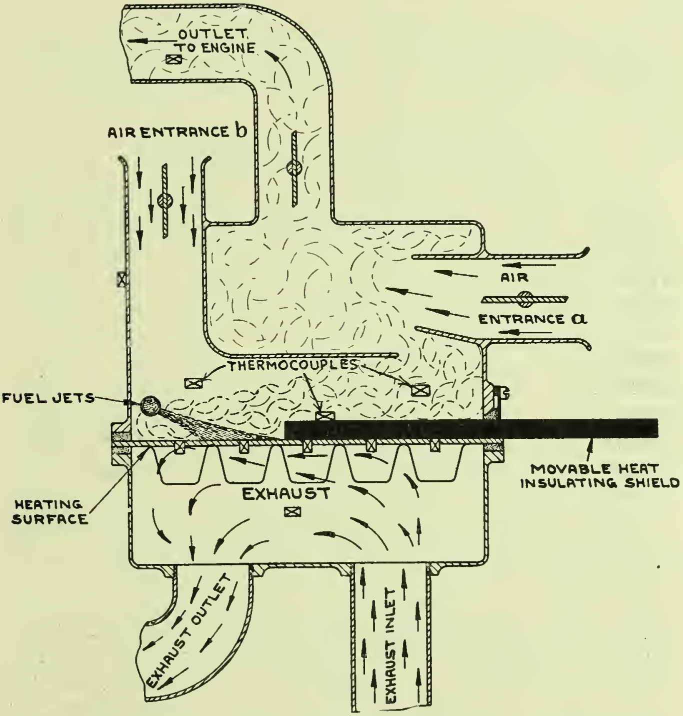
OUTLET TO ENGINE
AIR ENTRANCE b
AIR
ENTRANCE a
THERMOCOUPLES
FUEL JETS
MOVABLE HEAT INSULATING SHIELD
HEATING SURFACE
EXHAUST
EXHAUST OUTLET
EXHAUST INLET

FIG. 8—DIAGRAM SHOWING THE CONSTRUCTION OF THE EXPERIMENTAL FUEL HEATER USED

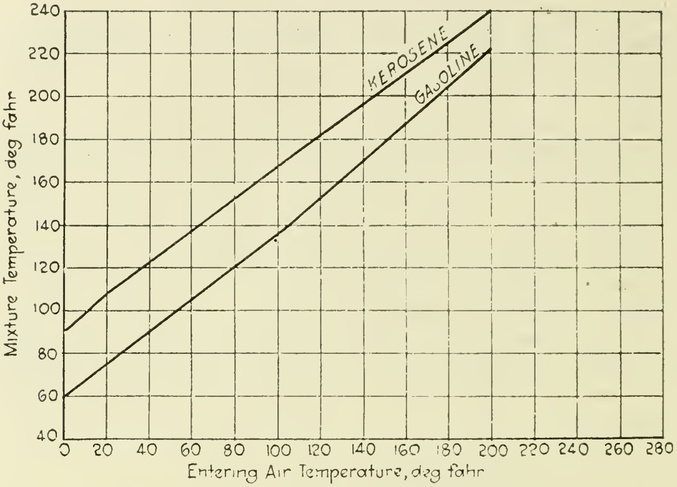

FIG. 9—APPROXIMATE TEMPERATURES BEYOND THE HOT-SPOT RESULTING FROM THE MIXING OF HIGH END-POINT GASOLINE ANE KEROSENE VAPORS, AT THEIR BOILING TEMPERATURES, WITH 15 TIMES THEIR WEIGHT OF AIR

NATURE OF ACTION AT HOT-SPOT

Our observations seemed consistently to show that whatever part of the fuel remains in contact with the hot-spot undergoes a sort of selective distillation, the light elements boiling off quickly and the heavier elements more slowly, being very much the same action as that in the distillation flask of Prof. R. E. Wilson's method of determining equilibrium solutions. Provided the temperature of the metal heating surface is above its boiling-point, each element of the fuel seems after boiling to depart from the pool of fuel on the heating surface as vapor at its own boiling-point. Very little heat is apparently communicated from the metal heating-surface and the liquid on the heating-surface to the airstream, and the final mixture-temperature is approximately such that its heat-content is the sum of that of the air part of the charge at its entering temperature and that of the fuel vapor at its average boiling-point. In other words, the heat balance and final mixture-temperatures with an exhaust hot-spot are substantially those obtained with the system shown in Fig. 4. This combination then results in a fog mixture, the temperature of which depends upon the boiling-point of the fuel, its specific heat, the mixture proportion and the temperature of the entering air.

Fig. 9 shows the mixture temperatures that should result from the combination of gasoline and kerosene vapor with varying temperatures of the entering air. It should be borne in mind that such low mixture-temperatures as these can scarcely be obtained under a motor-car hood because of preheating of the air, and later heating of the mixture, from sources external to the hot-spot. With the greatest care that can be taken, the mixture will enter the valve ports from 15 to 25 deg. fahr. hotter than indicated by the curves.

The temperature values given for motor gasoline and kerosene are those computed from the observations of Prof. R. E. Wilson and Daniel P. Barnard, 4th, described in their paper on Condensation Temperatures of Gasoline and Kerosene-Air Mixtures,[6] which check very closely our own observations. For convenient reference, curves of the heat-content of gasoline and kerosene at different temperatures and at pressures corresponding to those of the vapor in the mixture at full throttle, are given in Fig. 10 in terms of British thermal units and degrees fahrenheit. These also were obtained from the work of Professor Wilson and Mr. Barnard and can be used to

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[6] See THE JOURNAL, November, 1921, p. 313.

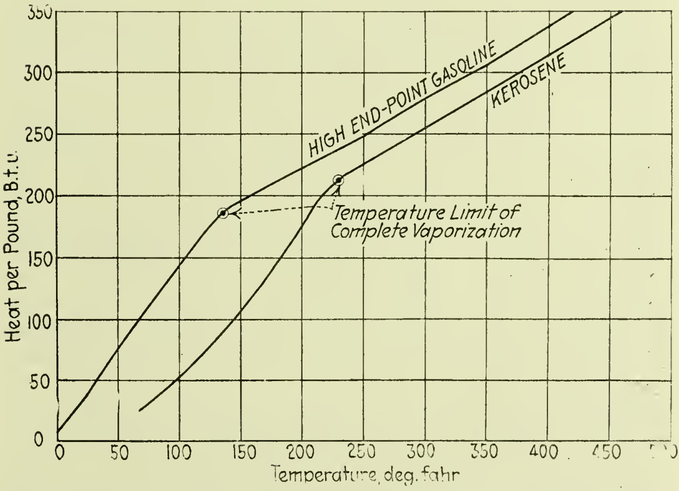

FIG. 10—SENSIBLE HEAT OF HIGH END-POINT GASOLINE AND KEROSENE AT VARYING TEMPERATURES IN A 15 TO 1 AIR-FUEL MIXTURE

compute the rise of temperature necessary with different mixture-proportions, initial air temperatures and the like.

With the ordinary hot-spot, the fuel remains on the surface only at low air velocities in the manifold, and in customary use a considerable portion of the fuel goes by into the engine without having gone through the process of evaporation and condensation, so that the conditions described in the foregoing, and in the paragraphs following, are only partly carried out. Such fuel as does not boil at the hot-spot is carried farther along into the intake-manifold where further evaporation takes place from its surface at a relatively slow rate.

When the temperature of the metal heating-surface was 200 deg. fahr. or more above the boiling-point of the fuel, there was a pronounced "spheroidal condition" and at times the drops of fuel would bounce around in the chamber like popping corn. Under this condition, a number of drops were caught up by the airdraft and swept out of the heating chamber into the intake-manifold without being evaporated. In our test, however, we found this only occasionally and as a temporary phenomenon, as it was only at the highest power output of the engine that the heating surface rose sufficiently beyond the boiling temperature for this condition to occur.

FINAL LIMITATIONS OF THE HOT-SPOT METHOD

With kerosene and with alcohol, the temperature of the metal surface sometimes fell below the boiling-point of the fuel. Under such conditions the evaporation took place by surface evaporation rather than by the combined surface and internal evaporation of boiling, and consequently a considerably larger surface area was necessary at this time. It should be borne in mind that, during boiling, the limitation of heat transfer was probably the ability of the ribs to collect heat from the exhaust, as the rate of transmission from the metal surface to the liquid was sufficiently rapid to take away the heat as fast as it was collected from the exhaust. When the evaporation is only from the surface, however, the extent of surface presented is the main limitation and it becomes necessary to spread the fuel out in a very wide film or to recirculate and respray it on the hot-spot. Whether this surface evaporation can be obtained is, in our estimation, the consideration that will determine how low we can go in the scale of fuel elements, using the hot-spot method of preparation. It seems very likely that we will not be able to use successfully, in general service, fuels the boiling-point of which lies above 500 to 550 deg. fahr., so long as a low "idle" is desired; because, even allowing for the effect of the reduced intake pressure in lowering the boiling-point, the exhaust temperatures will not be adequate.

The heat for evaporating the fuel is obtained from the exhaust through the mediation of the metal wall between, and the wall temperature is, therefore, lower than that of the exhaust but higher than that of the liquid film. The temperature of the heating surface, like the exhaust temperature, varies with the speed; but the percentage of variation is less. We found that the temperature drop of the heating surface under change of quantity or condition of mixture fed was a very good measure of the heat being taken up by the mixture. Any change of boiling-point, specific heat or latent heat of evaporation in the fuel is strikingly shown in the temperature of the heating surface. For instance, with the exhaust temperature at 1000 deg. fahr., the metal-wall temperature at one point was with motor gasoline 595 deg., with kerosene 535 deg., and with alcohol 175 deg. fahr. This emphasizes what has been implied previously, that with alcohol and kerosene it is very important that an adequate amount of surface be presented to collect heat from the exhaust.

We were very much surprised to find how little heat was taken up when air alone passed through the heating chamber. With a surface more than adequate to vaporize a full charge when the fuel was taken into the heating chamber, if air alone were passed through the heating chamber, the fuel being taken into the airstream beyond the heating chamber, the mixture temperature was only from 10 to 20 deg. fahr. above of that of the entering air and the engine ran very poorly indeed, with every evidence of poor fuel-distribution. Another evidence of this point is that, at a speed and load at which the temperature of the heating plate was 400 deg. fahr. with no circulation above it, when the air alone for the engine was taken over the plate, the plate temperature fell 30 deg. fahr.; but, when both the air and fuel charge were sprayed on the heating surface, its temperature fell 150 deg. fahr., or five times the temperature drop when air alone was passing. Observation of thermocouples at various points of the heating surface showed that the air received most of its heat in making the right-angle bend to flow across the metal surface, and that there was very little heat-transfer beyond the bend.

Since the liquid fuel on the heating surface is at no higher temperature than at its boiling-point, while the surface is hotter than this, it seems reasonable to believe that the communication of heat directly from the liquid fuel to the air might almost be ignored.

The test heating-chamber was constructed so that the area of surface exposed could be varied as necessary to maintain a constant mixture-temperature at varying speeds and loads. We were pleased to find that a constant area of surface was required for any given fuel, at both low and high speeds, at wide-open throttle; and that the area of surface which was right for wide-open throttle, was adequate for part-throttle running. An exception to this rule has already been noted in that, when the temperature of the plate fell below the boiling-point of the fuel, considerably more surface was needed.

Under the conditions of our test, the area of surface presented to the fuel, as found adequate for creation of a dry fog, is given in Table 2. For idling with kerosene and alcohol, a surface approximately 40 per cent greater than is given in Table 2 seemed necessary when the fuel was spread out in a thin film.

Under some conditions the surface collecting heat from the exhaust is a limiting factor of hot-spot capacity. In our experiments the surface presented to the exhaust was approximately three times that presented to the intake, giving the ratios of exhaust surface to piston displacement that are presented in Table 3.

The figures in Table 3 were obtained when there was a layer of soot about 1/32 in. thick on the exhaust surface. In their application to design it can be assumed that the whole mass of the exhaust-manifold around and adjacent to the hot-spot is effective in collecting exhaust heat and conducting it to the heating surface. This can be used as a guide to the number of ribs necessary. It

TABLE 2—ADEQUATE HEATING SURFACE FOR EACH SPECIFIED VOLUME OF ENGINE PISTON DISPLACEMENT

Kind of Fuel	Adequate Heating Surface, Sq. In.	Engine Piston Displacement, Cu. In.
Motor Gasoline	1	6.5
Aviation Gasoline	1	11.5
Kerosene	1	4.5
Grain Alcohol	1	5.6

TABLE 3—ADEQUATE EXHAUST SURFACE FOR EACH SPECIFIED VOLUME OF ENGINE PISTON DISPLACEMENT

Kind of Fuel	Adequate Exhaust Surface, Sq. In.	Piston Displacement, Cu. In.
Low-Test Motor-Gasoline	1	2.2
Aviation Gasoline	1	4.0
Kerosene	1	1.5
Grain Alcohol	1	1.9

is of course essential that there be an actual circulation of exhaust gases over the surfaces included in the computation.

Laboratory versus Motor-Car Heat-Conditions

It should be borne in mind that there was a very great difference between the temperature conditions of our laboratory test and those existing under a motor-car hood. We had jacketed the intake-manifold with asbestos and placed asbestos shields between it and the exhaust pipe so that there was no radiation of heat from the exhaust-manifold to the intake. We had a fan blast on the intake-system so that the temperature was that of the room, 75 to 85 deg. fahr., instead of the 140 to 160-deg. fahr. fan-blast temperature that, in summer, is ordinarily directed onto the intake-system under the hood; we also held the water temperature of the engine at 140 deg. fahr., which is somewhat lower than that of many engines when pulling a heavy load. The only heat applied to the mixture was at the heating surface.

When a hot-spot is applied to an intake-system that has a long extended surface in proximity to the exhaust-manifold, or one in which a large part of the intake-passage is jacketed in the cylinder-head, there is bound to be a very great difference between the mixture temperatures of summer and winter operation, although no greater difference than would exist in mixture temperature without the hot-spot. If, in addition, the air enters the carbureter at 140 to 160 deg. fahr. in summer, and between zero and 40 deg. fahr. in winter, it is obvious that some sort of temperature regulation will be needed. But it would seem logical to place the control where the variation occurs, on the hood temperature or on the air entering the carbureter, rather than on the hot-spot, the temperature transfer of which varies very little between all seasons and conditions of operation. Indeed, thus far we know of no completely successful effort for correcting for atmospheric and seasonal temperature-changes of the intake-system by change of the heat application at the hot-spot.

While on the subject of power loss from too much heat applied to the intake charge, it can be stated that the loss from expansion of the charge is less than is generally believed, and that the extreme and remarkable lack of power noted with some engines when there is too much heat on the manifold is due to a condition of detonation rather than the loss of so-called volumetric efficiency.

THE SEPARATING HOT-SPOT

It is well known that many of the shortcomings now experienced with present hot-spots are due to the fact that the fuel does not stay on their heating surfaces long enough to be subjected to complete vaporization and conversion to a fog. The result is that the operation of the engine is efficient and correct only above certain limiting mixture-temperatures. The obvious step seems to be to incorporate into the heating surface a separator that will catch the unvaporized fuel-drops and return them to the heating surface. We have found that very good results can be obtained with such a device, but it is essential that the separating hot-spot have *adequate capacity for transmitting heat from the exhaust to the fuel;* otherwise there will be a time when the fuel, although metered in the carbureter, will stop and collect in the heating chamber instead of going to the engine, exactly as fuel "loads" in our present intake-manifolds at low air-velocities. But with the current type of intake-manifold, if the supply of vaporized fuel is inadequate to run the engine, it can always be increased by using the carbureter mixture control and raising the engine speed to a point where the air velocity carries a firing charge to each cylinder. With the separating hot-spot, this cannot be done, at least not until the separating chamber is filled with liquid fuel, but it is remarkable how well a passenger-car engine will perform on fuels so heavy that they cannot be vaporized and passed on to the engine except at part throttle. For truck and tractor use, in fact for all heavy-duty work, the separating hot-spot, if properly designed, presents the great advantage of absolutely preventing crankcase and cylinder-wall lubricant-dilution.

Successful application of the separating hot-spot demands only an ordinary knowledge of the laws of physics relating to heat, and presents much less difficulty than a number of other problems that our automotive engineers have solved. It is only a question of getting an adequate heat-supply from the exhaust and of excluding heat communication from other sources. As suggested in the foregoing, the entrance-air temperature should be the lowest that can be obtained, and care should be taken to avoid conduction of heat to the intake system beyond the hot-spot. In particular, careful attention must be given to the heat insulation between the heating surface and the remainder of the enclosing walls of the intake-system, as a tremendous amount of heat can be conducted across the ordinary flange-joint.

Our experience indicates that the separating hot-spot should always be located in the main exhaust line and that it is hopeless to attempt to pipe the exhaust across a T-head engine, or the like, as the temperature drop will result in too low an exhaust temperature at the lower speeds. Also, the actual flow of exhaust to the hot-spot will be a function of the muffler back-pressure, which will result in exaggerated temperatures of the mixture at high car-speeds.

NATURAL VARIATION OF AIR TEMPERATURE UNDER THE HOOD

In the quantitative computation of heat transfer, we first must take into account the very wide variations of temperature that the air charge and fuel supply undergo before they enter the intake-manifold system, on account of the large range of variation of hood temperature. Fig. 11 is given as a rough indication of the various changes in temperature that a molecule of air undergoes in getting from the external atmosphere to the cylinder port, without purposeful application of heat to the intake charge, other than the commonly used hot-air stove around the exhaust-pipe. Starting from atmospheric temperature, the temperature of the air is raised between 30 and 60 deg. in passing through the radiator. It has been our observation that there is a greater difference between the motometer temperature and that of the external air in summer than in winter. Perhaps some of the radiator engineers can tell why this is so. In the summer there is sometimes an additional rise of temperature under the hood due to the radiation of heat from the engine. The rise of temperature from the hot-air stove is presumably about the same in the summer and in the winter but on many cars an appreciable portion of the heat added by the stove in the winter is lost before it gets to the carbureter, because of the cooling effect of the fan-blast of relatively cold air on a long length of flexible tubing. The temperature-drop in the carbureter and the manifold due to vaporization is indeterminate, depending upon the fuel, the temperature and the vacuum. In many cars the intake-manifold is so close to the exhaust that under full load the temperature is raised considerably by the cross radiation. We have sometimes gained 3 to 4 hp. in a maximum of about 70, by cutting off this radiation with asbestos board.

Fig. 11 will give an idea of the range of natural temperature-variation with which our intake-systems have had to deal. Between the temperature of the air entering the intake-system just after starting in the winter and that during a long run in the summer, there easily may be a difference of 120 deg. fahr. Very few current applications of heat to the intake charge, by either hot air or hot-spots, affect the temperature one half this amount. Any effort to attain minimum charge-temperatures in actual practice must include means for dealing with the natural temperature-variation under the hood.

MEANS OF TEMPERATURE CONTROL

With heating methods that approximate Fig. 1, the heating surface should perhaps be in two sections, one of which is in action at all times, while the other may be thrown open to the exhaust, either by a seasonally regulated valve, or by the dash mixture-control of the carbureter.

Arrangements such as that shown in Fig. 2 can be controlled within certain limits of temperature by using a hot-air stove on the exhaust line that has at least three times the heat capacity of those in common use today, with a valve adapted to cut-off part of this hot air and admit cold air as the engine warms-up. The regulation should preferably be automatic.

In the methods of Case No. 2, Figs. 4 and 5, no particular regulation for variation of atmospheric temperature is necessary. The heating action is almost independent of the outside temperature. With this type of construction, I have always recommended making the heater large enough so that cool air from outside the hood can be taken into the carbureter in the summer time.

Fig. 6, like Fig. 1, would perhaps be taken care of best

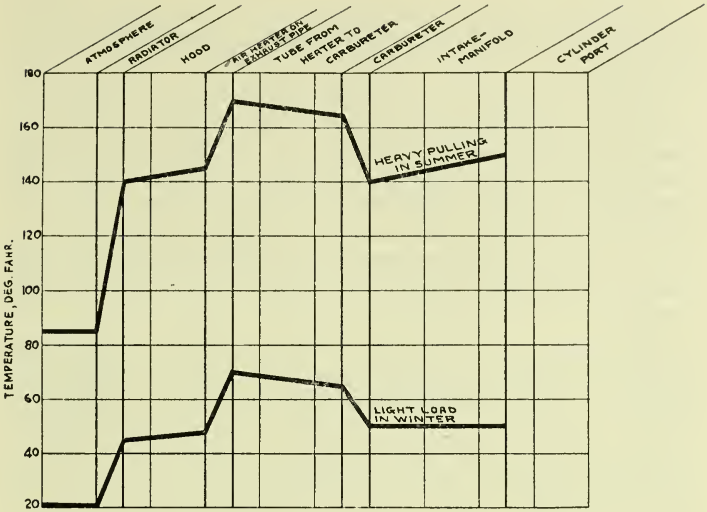

Fig. 11—Curves Showing the Range of the Natural Temperature Variation of the Air Charge

by a regulable variation in hot-spot area. Difficulty is experienced in practice in confining the heat to the region where it is desired. In warm weather the heat from the warm hood atmosphere tends to conduct across the flange junctions and through the walls of the heating chamber. In future designs we may find thick heat-insulating material, or spacers of refractory tile, used to separate the hot from the cool portions of the heater.

Homogeneous Mixture Quality

As has been brought out in the foregoing, a homogeneous mixture requires a fine spray from the carbureter issuing directly into the heating region. If the fuel is allowed to condense or gather on the walls, it will reach the hot surface in waves and irregularly timed splashes, under which conditions the carbureter setting must always be somewhat rich, and many details of engine operation will suffer. Acceleration is always more difficult when there is a fuel lag between the carbureter and the heating surface.

The arrangement, common in many heavy-duty engines, of locating the governor between the carbureter and the hot-spot, is very bad. Everything indicates that the carbon deposit will be reduced to the minimum and crankcase-oil dilution eliminated only when this custom is disregarded and the carbureter is placed close to the hot-spot.

Several 1922 engines that have the property of operating very smoothly on extremely lean mixtures, have intake-manifolds that are characterized by a hot-spot at the carbureter opening and additional contact with the exhaust-manifold a little farther along, usually at a point of division of the fore-and-aft reaches. Apparently the second "spot" catches some of the particles that elude the first one and gives a more complete and steady evaporation.

One important requirement of a successful fuel-charge heater is that it should warm-up and get underway quickly. The walls should be thin, and, if cast, should be lightly ribbed on the exhaust side. Aluminum combines low specific heat and rapid conductivity and is a very suitable material for a cast hot-spot heating surface, if there are no shielded parts that will become heated to such a degree as to melt.

PREVENTING LIQUID FUEL REACHING THE CYLINDER

It is recognized generally that it is desirable to prevent liquid fuel from reaching the cylinder and it has been claimed for many designs that they have this action. We have tried models of a number of them and have found that few impede the travel of liquid fuel to the cylinders in even a slight degree. With transversely ribbed elbows, for instance, the fuel drops are caught off the tips of the ribs by the air eddies and snatched through the elbow as if no ribs were present. This, of course, is with air velocities above 70 ft. per sec., and part of the lively action naturally is due to the spheroidal condition already described.

We have used centrifugal force, surface tension and the force of gravity to separate the unvaporized drops. Careful combination of all seems to be required to achieve complete separation. A partial separation, which should be very effective at low engine speeds, can be obtained by abruptly increasing the manifold area above and beyond the hot-spot. This would allow the heavy drops to settle down and again be hurled against the heating surface. The separation and recirculation would obviously be beneficial to the action of either Figs. 4 or 5, but *the heat supply must be adequate or the fuel will not reach the engine,* with an actually functioning liquid-fuel separating device.

In our work with various types of fuel heater, we have experienced a slight but important change of viewpoint, perhaps a keener realization of the truth, in the problem of supplying fuel to internal-combustion engines. This I would like to communicate to the Society. After watching the fuel, in an accumulation equal to many cylinder charges, bubbling, splashing, sometimes lying quiescent on the heating surface of glass-walled hot-spots, and sometimes swept through in a high-velocity spray, one fact stands out: the *rate of fuel-feed from the manifold to the cylinder* primarily governs the conditions of combustion, and the rate of fuel-feed to the manifold is an indirect rather than a direct controlling factor, as regards the mixture proportion of the charge actually used by the engine.

This point of view, we believe, is the proper one from which to consider the problem of the efficient use of fuel in our engines of today.

MORE CAR-MILES PER GALLON OF FUEL[1]

By O C Berry[2]

Economy tests carried out in France indicate that it is possible to obtain a larger number of miles per gallon from cars made there than from cars made in this Country. The author states that it would be well to make a careful study of the factors influencing car economy and to assure that our future car models take full advantage of all possible means of increasing their economy.

Figures are presented showing the extent to which economy can be increased by changing such factors as the carbureter adjustment, time of the spark, rear-axle ratio and speed of driving. A car that normally will go 21 miles per gal. under favorable test conditions at 20 m.p.h. was increased to 43 miles per gal. at 20 m.p.h. The study is not complete but has gone far enough to demonstrate its value. This progress report is presented to stimulate thought.

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[1] Detroit Section paper.

[2] M.S.A.E.—Chief engineer, Wheeler-Schebler Carbureter Co., Indianapolis.

The American automobile has been developed to a stage where it is a remarkably reliable vehicle. The hundreds of different makes of car do not vary in economy much more than their differences in weight would account for, and the offhand conclusion logically would be that there is probably not much more room for improvement in economy than in mechanical reliability. In this connection the recent economy tests in France are of interest. A Voisin limousine weighing 5300 lb. ran 28.3 miles per American gal. A Citroen car weighing 2500 lb. traveled 53 miles per gal., and a Petit-Peugeot car, weighing 1200 lb., 76.9 miles per gal. Contrasting these figures with the performance of the average American car makes it evident that the margin for improvement is indeed great. Probably no one feature of American cars has received less careful attention from automotive engineers than the capacity for economical running. No fuel-economy features are incorporated in the European cars mentioned that we do not know about and understand, and I do not concede for a moment that the foreign engineer is one whit more resourceful or well informed than the American. It is therefore incumbent upon us to look the situation squarely in the face and, recognizing the importance of fuel economy, to study carefully all of the factors influencing the number of car-miles per gallon of fuel, and to make certain that every possible improvement is included in the design of our future models.

CARBURETION

One of the outstanding reasons for the poor mileage of the average American car is poor carburetion or, more accurately, poor carbureter adjustment. To make this clear, it will be necessary to show the effect of the richness of the fuel-mixture on the power and economy of an internal-combustion engine. In studying this point, use was made of a Willys-Knight engine mounted on an electric dynamometer. The results of these tests are shown by the curves in Fig. 1. In these curves the number of pounds of gasoline per pound of dry air in the fuel-mixture is plotted horizontally and brake horsepower and percentage of thermal efficiency are plotted vertically. In carrying out the tests the throttle was arranged so that it could be securely fastened, and all of the tests were run at a constant engine speed. The gasoline was weighed to 0.01 oz., the air metered to 0.10

cu. ft. and the brake-load, speed and temperatures were carefully measured. Under these conditions and with the carbureter adjusted to give a rich and powerful mixture, the first test was run. The power, efficiency and mixture ratio were then computed, and a point established on both the power and the efficiency curves. The gasoline adjustment was then made slightly leaner, the brake-load adjusted to produce the correct speed, and a second test was run. This procedure was repeated until the mixture became too lean to allow proper engine performance. More gasoline was then introduced each time until the mixture was entirely too rich. The mixture was thus made alternately richer and leaner until a suf-

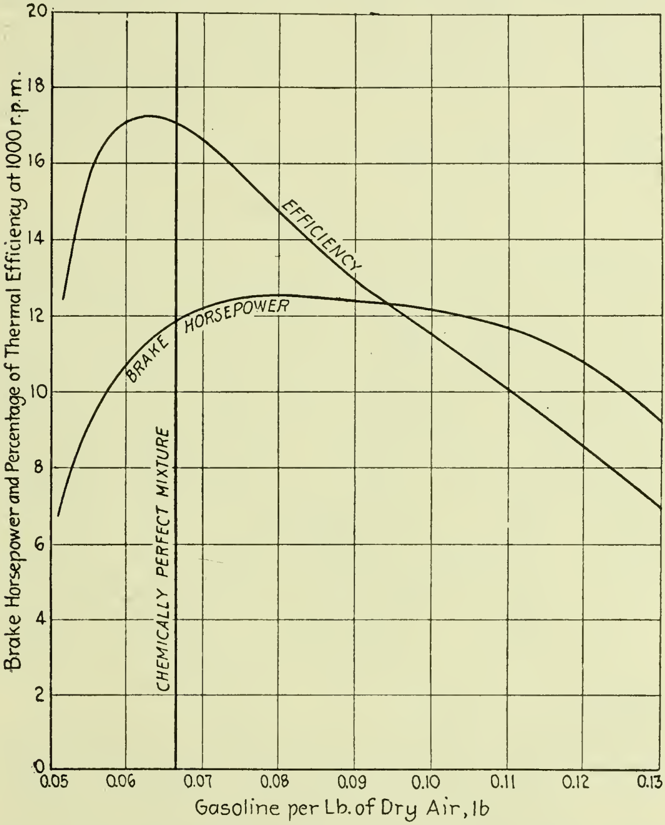

FIG. 1—CURVES SHOWING THE RESULTS OF TESTS TO DETERMINE THE EFFECT OF THE RICHNESS OF THE FUEL-MIXTURE UPON THE POWER AND ECONOMY OF AN INTERNAL-COMBUSTION ENGINE

ficient number of points had been located on each curve to indicate clearly the effect of the richness of the fuel-to-air mixture on both the power and the efficiency of the engine.

It will be noted that there is a wide range of mixtures through which the richness has little effect on the power of the engine. The lean and the rich mixtures give equally good power, and both result in what seems to be perfect engine performance. This accounts for the fact that the comparatively crude carbureters of earlier days gave mainly satisfactory performance.

The range of mixtures giving the highest efficiency is very narrow, and corresponds to about the leanest mixture with which the engine will run without missing. As more gasoline is added, the efficiency drops off very rapidly; the richest mixture producing full power will result in only about one-half of the maximum efficiency. This explains why it is that of two cars of the same make performing nicely, one may go nearly twice as many miles per gallon as the other.

WHY OVER-RICH MIXTURES ARE USED

There are several reasons why nearly all of the carbureters are adjusted so as to give too rich a mixture when the engine is hot. One is that gasoline flows much more slowly when cold than when warm; an opening large enough to deliver the required charge to a cold engine will inevitably deliver too much when the engine is warmed-up. Another is that cold air is more dense than warm air; the weight of air delivered through the air-opening in a carbureter is greater when the air is cold than when it is warm, thus causing the mixture to become richer as the temperature of the engine increases. Thus, the mixture furnished by a carbureter with a fixed adjustment will become richer as the temperature rises. To make matters worse, not nearly all of the gasoline in a cold mixture is vaporized so that it can be burned. This makes it necessary to supply a considerable excess of fuel in a cold mixture to get the engine to run at all.

The American public has been taught to demand a car that will start easily in any kind of weather and continue to run without requiring any attention or adjusting. They have been supplied with a large variety of non-adjustable, or what has aptly been termed "foolproof," carbureters. These carbureters will deliver a mixture rich enough to start well in cold weather and, due to the great range of the explodable mixtures of gasoline and air, will seemingly keep the engine running perfectly with full power, in summer or in winter. The only fault to be found is that, when the engine is hot, the mixture is much richer than is necessary and the number of miles per gallon is correspondingly low. A non-adjustable carbureter cannot be made so that it will make starting easy and also give high efficiency in driving.

A carbureter adjusted for maximum economy on a hot engine will deliver too lean a mixture to start with when the engine is cold. There are two ways in common use for meeting this situation. One is supplying a choke to shut-off the main supply of air to the carbureter, thus enriching the mixture. The other is providing a means of adjusting the carbureter from the dash or the steering-post of the car.

The action of the choke is very severe. It causes raw gasoline to be sucked into the engine in large quantity. Its action becomes increasingly severe as the engine speeds up; if the mixture is rich enough to start the engine it will increase in excess fuel as higher speeds are reached. This makes it necessary to readjust the choke every time the speed is changed.

The dash adjustment can be made very satisfactory in action if it is used correctly, and convenient as well. Its use is easily understood, because the idea is to get it set as lean as possible; it is very easy to tell when it is too lean, for the engine will lose power and backfire through the carbureter. A proper dash adjustment set right for one speed and load will be correct at all speeds and loads at the same temperature, and will need to be changed only as the temperature changes. The mixture should be just right for a hot engine with the dash adjustment set at its leanest position.

Discussion of Curves

The curves in Fig. 1 show that a richer mixture is required for maximum power than for maximum efficiency. It is therefore obvious that a carbureter adjusted for maximum miles per gallon will deliver too lean a mixture to give the full power of the engine.

This difficulty can be obviated in the following manner. When driving on a level road at any speed the law allows, the engine is never called upon to deliver nearly its full power. Under these conditions the most important consideration is economy. When a steep hill is to be climbed, acceleration is desired; if a neighbor wants to race, full power is the important thing. Constant-speed driving is done with the throttle pretty well closed, and the power work with the throttle wide-open. It is possible to design a carbureter that will furnish the engine with the most economical mixture at all ordinary speeds on a hard level road, and with the most powerful mixture when the throttle is wide-open.

The fuel is almost never entirely vaporized in the intake-manifold of the engine. A considerable portion flows along the manifold walls as a liquid, lagging behind the air that passed through the carbureter with it. When the engine is idling this layer of liquid is very thin, but under full load it usually forms a fairly large stream. When the engine is put under load quickly, the air rushes ahead and leaves at least a portion of liquid on the manifold walls. If the mixture is lean enough to be efficient under steady running conditions, this temporary impoverishment will often be sufficient to stall the engine. It is therefore necessary to incorporate in the design of the carbureter some special means of temporarily enriching the mixture during acceleration. There are many types of construction in use, some of which are very satisfactory. It is therefore possible to obtain perfect acceleration, even when using a mixture lean enough for maximum efficiency. The more perfect the manifold design is, the easier this becomes and, with the very best manifolds, very little special provision is necessary for acceleration.

THE IDEAL CARBURETER

In my estimation the highest type of carbureter yet developed for American gasoline will

(1) Furnish the engine with the most efficient mixture when the car is driven at a constant speed on a hard level road
(2) Provide the engine with the most powerful mixture when the throttle is wide-open
(3) Make perfect acceleration possible, even when adjusted for maximum economy
(4) Have a dash adjustment that will make starting and warming-up an easy matter, even in the coldest weather

There are carbureters on the market that meet all of these requirements. They make a greatly increased number of miles per gallon possible, and are more satisfactory to handle than the cruder fool-proof varieties.

THE TIMING OF THE SPARK

The second factor influencing the economy of the automobile is the timing of the spark. If the spark passes too early it will lessen the power of the engine and tend to produce what is called a "spark knock." If it passes too late it will also reduce the power of the engine. The proper timing of the spark in any given engine will vary with two things, the load the engine is carrying and the speed at which it is running. Nearly everyone knows that the faster an engine runs the earlier the spark must pass to give best results. The load condition has received comparatively little attention, and I will therefore enlarge upon it.

In making a study of this point an engine was mounted on an electric dynamometer in such a way that the exact time of the passing of the spark could be read, as well as the speed of the engine and the brake-load carried. The engine was run through a series of different spark-settings at each of a series of different throttle-openings. The results of these tests are shown in Fig. 2. In the curves the lead of the spark in degrees on the flywheel is plotted horizontally, and the brake-load carried is plotted vertically. All of the tests were made at 1000 r.p.m. Each curve was made at a constant throttle-opening, the brake-load being changed along with the spark-timing to keep the speed constant.

It will be noted that a lead of 20 deg. is required for maximum power at full throttle, and that a lead of 25 deg. will result in a knock. As the throttle is closed gradually, the lead required for highest power is increased until, with the lightest load carried, a lead of 40 deg. is required. At this throttle-opening the engine will carry only 60 per cent as much load with a 20-deg. spark-lead as with a 40-deg. lead. It is so disagreeable to have an engine knocking during acceleration and hill-climbing that the spark is almost never advanced beyond the point where knocking occurs at full load. Since the engines in our American cars run at such a low percentage of their power capacity, the spark is timed much later than it should be for greatest economy, which results in a considerable loss in miles per gallon.

The vacuum in the intake-manifold of an engine varies in almost inverse ratio with the brake-load carried. It therefore follows that the lead required by the spark at a light load, beyond that required at full load, will vary directly with the intake-manifold vacuum. This fact has been made use of in producing an automatic device for changing the time of the spark to correspond to the load carried. When properly installed, together with a standard controller compensating for speed, this device provides a completely automatic and very accurate means of timing the spark. This device is not on the market at present, but I have conducted a series of tests showing its merit and feel confident that it will be perfected and presented to the public in the near future. When properly installed, it should add considerably to the economy obtained by the average driver.

The speed at which a car is driven will cause a considerable variation in the economy obtained. This is shown in Fig. 3. The tests here reported were made on a hard level road, running in both directions over a carefully measured course. The car was tested with the top

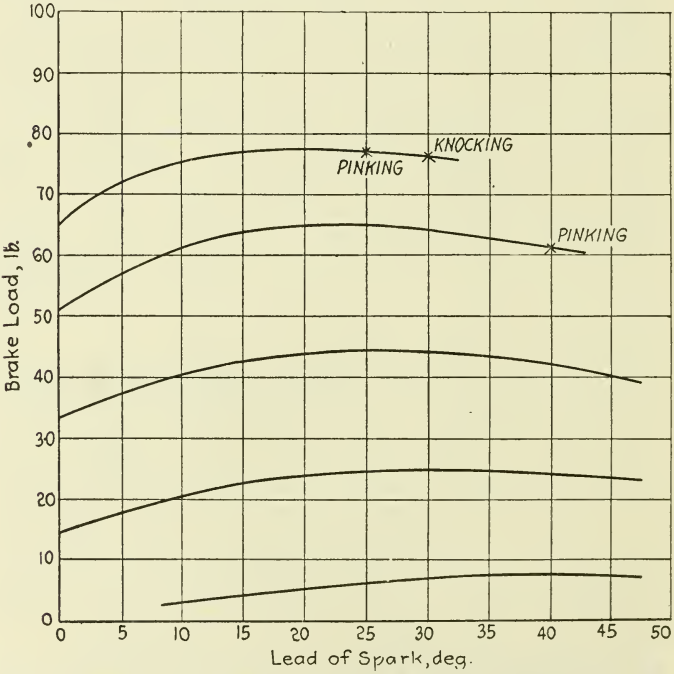

FIG. 2—RESULTS OF TESTS TO ESTABLISH THE RELATIONSHIP BETWEEN THE LEAD OF THE SPARK AND THE BRAKE-LOAD

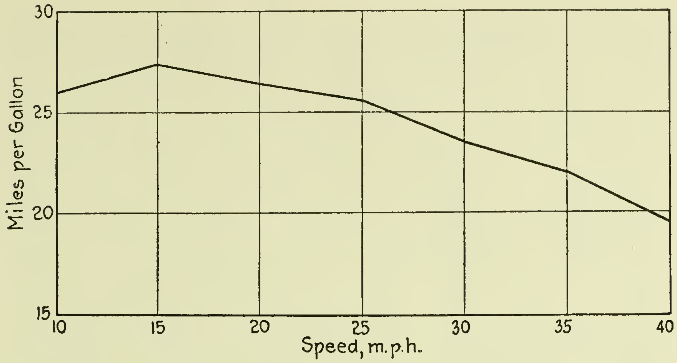

Fig. 3—Curve Showing How Changes in the Speed at Which a Car Is Driven Affect the Mileage Obtainable from a Gallon of Gasoline

and windshield up and at a weight of 3100 lb., including the driver and the observer. Cars of this make probably will average about 21 miles per gal. under test when in the condition in which the average driver keeps his car. By freeing the brakes, having the spark accurately timed and the carbureter carefully adjusted, the record indicated by Fig. 3 was obtained.

The Load on the Engine

Another opportunity to improve the economy of our automobiles has to do with the large reserve power in the engines. A good American car can climb a steep hill in high gear, and accelerate very rapidly. This delightful "activity" necessarily means that under ordinary driving conditions on a hard level road the engine is called upon to exert a very small percentage of its full torque capacity. To get accurate information on this point, a car was chosen of a make known to be carefully designed, well built and having a good average accelerating ability of about 3.2 ft. per sec. per sec. Tests were made showing the torque capacity of the engine and the power required to propel the car at varying speeds. The results of these tests can be taken as representing the average performance of good American cars.

The method of conducting the tests was to insert between the carbureter and the intake-manifold a plate having a hole drilled through it that would serve as a throttle-opening. A determination was then made of the maximum speed the car could attain on a long stretch of hard level road with this orifice in place. This test was repeated with a number of orifices of different sizes. With the engine removed from the car and connected to an electric dynamometer, tests were run to determine its torque capacity at varying speeds with each of the orifices in place. Knowing the size of the rear wheels and the gear-ratio, the engine speed corresponding to any car speed is accurately known. The torque required to drive the car at any one of these maximum speeds for a given orifice is the torque capacity of the engine at that speed with that orifice, as shown by the dynamometer tests. The results of these tests are shown in Fig. 4. In these curves the engine speed is plotted horizontally and the brake-load at a radius of 15.75 in. is plotted vertically. The upper curve shows the torque capacity of the engine and the lower one shows the torque, measured at the flywheel of the engine, that is required to propel the car at corresponding speeds. The distance between the upper and lower curves is therefore proportional to the reserve power of the engine, and represents its ability to accelerate rapidly or climb a steep hill.

In this particular car an engine speed of 1000 r.p.m. corresponds to 20 m.p.h. It will be noted that at this speed only about 14.7 per cent of the torque capacity of the engine is used when driving on a good road. Anyone familiar with the performance characteristics of an engine knows that this fact alone will account for a large reduction in the thermal efficiency obtainable from the engine under ordinary driving conditions. This is such an important consideration that a special series of tests was carried out to show the exact situation.

EFFICIENCY AT DIFFERENT LOADS

It is well known that the brake thermal efficiency of an engine varies from zero at no load to a maximum that occurs at a little less than full load. Engines differ according to their design in ability to perform under different conditions, some doing comparatively better at light loads and others at full load. Tests were therefore carried out on the engine previously used to determine the effect on its thermal efficiency of changing the load.

The engine was installed on the dynamometer and run at 1000 r.p.m., at a series of different throttle-openings. At each throttle-opening a series of tests was run at different carbureter settings, similar to the tests plotted in Fig. 1, and showing the mixture corresponding to highest-efficiency at that throttle-opening, together with the corresponding power and efficiency. The brake-load was expressed in terms of brake mean effective pressure, and a curve drawn through the points showing the highest efficiency at each load. This curve, shown in Fig. 5, indicates the maximum efficiency obtainable with this engine at each brake-load, the engine running at 1000 r.p.m. The efficiencies will vary as the speed is changed, so that this curve should be interpreted as applying only to the one speed.

The carbureter was adjusted leanly enough to give maximum economy at all loads until the throttle was wide-open. After this point was reached, the increases in power had to be obtained by enriching the mixture, until the full power was obtained. These richer mixtures result in reduced economy, thus causing the curve in Fig. 5 to show a decided drop as full power is approached. This curve therefore offers one more illustra-

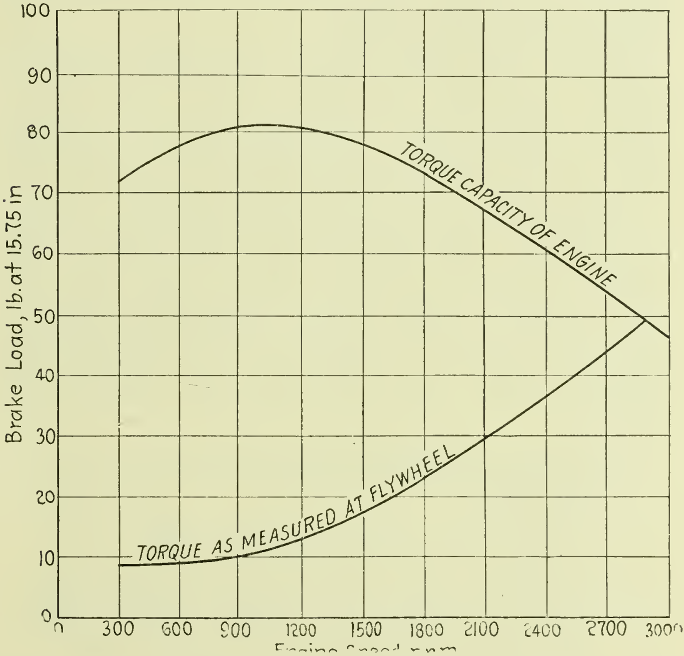

FIG. 4—RESULTS OF TESTS MADE TO OBTAIN THE TORQUE REQUIRED TO DRIVE THE CAR AT VARIOUS ENGINE SPEEDS AND THE TORQUE MEASURED AT THE FLYWHEEL OF THE ENGINE

tion of the importance of using lean mixtures when high economy is desired.

Fig. 5 shows how important it is to run the engine under a comparatively large percentage of its full-load capacity. Since the miles per gallon of any car under given running conditions will vary directly with the thermal efficiency of the engine, a change in the load factor on the engine can result in greatly increased economy. This fact was checked in part by installing a different rear-axle ratio and noting the change in the car economy. The standard ratio was 4⅞ to 1, and resulted in a load factor of 14.7 per cent at a speed of 20 m.p.h. Under these conditions the car could go 28 miles per gal. when loaded so as to weigh 3100 lb. A gear ratio of 2½ to 1 would result in nearly twice the former load factor, and Fig. 5 shows that this should result in an increase in fuel economy of about 50 per cent, or to 42 miles per gal. Under test the car actually ran 43 miles per gal. These tests were all made with a carbureter equipped with a dash adjustment to make starting easy in cold weather.

Efficiency and great ability to accelerate cannot be obtained at the same gear-ratio. The former requires that the engine run at a high percentage of its torque capacity and the latter the other extreme. We cannot afford to compromise the delightful activity of our cars. On the other hand, I feel confident that the first company to produce a car having the activity to which we are accustomed and capable also of going more miles per gallon than any other car of its weight in the Country, will become both rich and famous. We can accomplish this if we have four speeds forward. Third speed could be called the "power" or "speed" gear and the fourth the "economy" gear. Such an arrangement would be beneficial by

(1) Greatly increasing the miles per gallon
(2) Reducing the engine vibration at the common touring speeds
(3) Increasing the speed at which one could tour with comfort and satisfaction on good roads
(4) Enabling one to tour at high speed with the mental satisfaction that would result from knowing that one is not punishing the engine, while any other car at the same speed would be going entirely too fast for its own good
(5) Greatly reduce the wear on the engine, thus prolonging its life and reducing maintenance costs

The disadvantages are that (*a*) the four-speed transmission would be expensive when sufficiently well-made to run quietly on two gears and (*b*) the gears would need to be shifted more often.

The possible advantages are therefore fundamental and far-reaching and the disadvantages of a practical nature and, although difficult to overcome, not too difficult for automotive engineers when they really become interested in mastering them.

EDUCATION OF THE BUYING PUBLIC

The automotive engineer gives more attention to designing a car that will sell well than to making it meet his ideas of perfection. Our fuel has been cheap and plentiful and economy has not been demanded by the

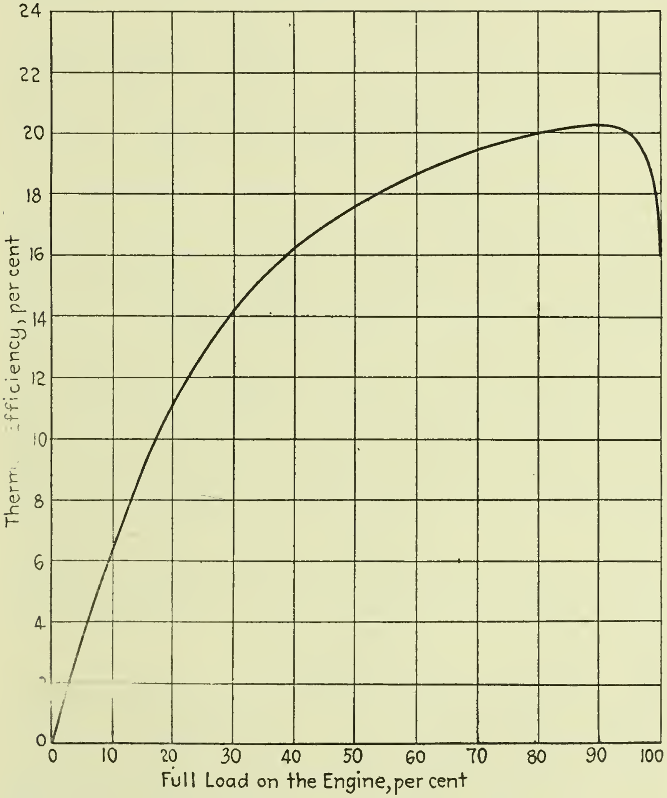

FIG. 5—VARIATION IN THE THERMAL EFFICIENCY OF THE ENGINE AT DIFFERENT LOADS

average buyer. He has rather been taught to demand a car that will accelerate well and climb any hill on high gear. He seldom stops to think how much the activity of his car costs him in gasoline, oil and repairs. He has never been educated to see how much can be gained by accepting different standards of performance. The automotive industry should make it a point to get this information before the public and thus create a demand for a more economical car.

The driver is not aware of how much fuel his car is using as he rides along and is accordingly not apt to handle it economically. For this reason it will be well for us to give the flowmeter serious attention. Such an instrument has great possibilities as a means of educating the public to demand greater economy in their care and would logically result in cars being kept in better mechanical condition. If a flowmeter can be developed that will never cause trouble, it should be installed as standard equipment on all new cars. It has already been developed to a point where it can unquestionably be made of value to the service-man.

Economy Contests in America Desirable

It might be well to run a series of economy tests in this Country similar to those that have been carried out in France. Unfortunately, a number of American engineers have questioned the practical value of these tests. They state that the carbureters are adjusted so leanly in these tests that the cars are incapable of good general performance and would be unfit for general use without changing the carbureter adjustment. As a matter of fact, it is possible to build a carbureter that will perform well even when adjusted for maximum economy, and these tests would prove this. In planning such contests a careful set of standards of performance should be established. These standards should be high and every car required to meet them before taking part. Under these conditions the results would be of real practical value and lead to wonderful development in the way of a public demand for an economical car.

By applying the ideas brought out in this paper to a standard representative car, its mileage was increased from 21 to 43 miles per gal., under similar test conditions. Our knowledge of how to increase car economy is not yet complete. There are factors influencing miles per gallon that are not mentioned here. This paper is merely a progress report presented at an early stage. I feel however that it has gone far enough to demonstrate the value of the study and warrant a conviction that American engineers can be depended upon to produce cars showing economies that now seem almost impossible.

By lessening wastes in running and storing petroleum, by perfecting the processes of making gasoline and especially by developing ways of producing substitute fuels economically and in large quantities, the chemists of the Country can help make it possible for future generations to ride in automobiles. In the meantime, automotive engineers should be steadily increasing the economy of our cars and thus add to the value of each gallon of available fuel.

THE DISCUSSION

W. S. JAMES:—Regarding the comparison Mr. Berry made between American and foreign-built cars, it should be borne in mind that the European tests he quoted were run under very special conditions. When driving around the curves, if a driver slowed-down very much his engine usually stopped. All the cars were adjusted for very economical running to obtain the maximum number of miles per gallon of fuel. The Petit-Peugeot car weighed about 1000 lb. and gave a performance of 70 miles, or 35 ton-miles, per gal. Let us compare this with the Buick performance. The Buick weighed at least 3000 lb. and gave 20 miles per gal., or about 30 ton-miles per gal. There is little doubt that, under special conditions similar to those existing in the tests discussed, the Buick performance could be increased to 25 miles per gal., which would mean about 37½ ton-miles per gal. or a figure at least as good as that of the Petit-Peugeot.

I had an opportunity in Europe to obtain manufacturers' statistics on the fuel mileage of European cars and compared them on the basis of ton-miles per gallon with similar statistics from American companies exhibiting cars at the New York Automobile Show. The figures can possibly be taken as equally optimistic in both cases. The results of the comparison showed that the number of ton-miles per gallon obtained from cars in France was no greater than the average number of ton-miles per gallon obtained in the United States. In England the number of ton-miles per gallon is possibly 10 per cent greater than in the United States. In both instances, however, the average number of miles per gallon is greater by possibly 20 per cent in England and from 10 to 15 per cent in France. This simply means that we are building heavier cars, with engines of at least the same efficiency.

Using the volume of air drawn into the engine per ton-mile as another basis of comparison, American engines draw in about 30 to 35 per cent more air per ton-mile. This means that our engines run at more nearly closed throttle, and, as Mr. Berry pointed out, our pumping losses are very much greater. In spite of the fact that our engines pump 30 per cent more air, American cars are still only 10 per cent low in ton-miles per gallon. This does not mean that we are any further behind European performance in actual engine efficiency, but that we are using heavier cars. In fact, at the New York Automobile Show, I could find only about four cars made in the United States that weighed less than 2000 lb., loaded. In England there are about 65 makes of car that weigh over 2000 lb. and 67 that weigh under 2000 lb.

If the amount of gasoline burned in internal-combustion engines is 4,000,000,000 gal. per year, and it were to flow over Niagara Falls at the mean rate of flow of the Niagara River or 222,000 cu. ft. per sec., it would take a little more than 40 min. to flow over the falls. The total volume of crude oil from which this gasoline was produced would require over 2½ hr. to flow over the falls. This helps us to understand what a gain it would be to save only 10 per cent of that volume of liquid. Even a 1-per cent saving in gasoline on the average amount used in this Country would mean a large volume of liquid.

I thought I knew how to drive a car economically, but since I have had a gallons-per-hour flowmeter on my car I have changed my mind. I think Mr. Berry's point about the use of flowmeters was stated very well. The greater part of the waste of gasoline is not through the ignorance or the lack of application of the knowledge of the engineers, but through a lack of knowledge and information on the part of drivers all over the Country. If a driver had some sort of tell-tale that would indicate whether he was using more gasoline at one time than at another it would help to save gasoline. The instrument could be graduated to indicate dollars and cents per hour or miles per gallon, or whatever seemed desirable.

O. C. **Berry**:—I do not wish to convey any unfavorable impression regarding American engineers or car performance in this Country. Some tests, such as Mr. Nelson's at Indianapolis, show performances considerably better than those of the French. But the performance of our cars in the hands of the average driver can be improved. The problem we are facing is to persuade the average driver to increase the number of miles he obtains from a gallon of gasoline.

P. F. **Howell**: — The flowmeter would no doubt be better understood if it were called a gasoline speedometer. It is an instrument that will indicate the amount of gasoline that is being consumed per hour when used in connection with an automobile, and gives the miles per gallon delivered by the car when read jointly with the speedometer. The instrument was developed about a year ago and was found to be of value when used in the final adjustment of new cars before delivering them. The volume of the distributor's business comes from the small contracts for from 5 to 20 cars each. This business is often rather expensive in regard to service and replacements of parts; the only possible way the necessary evil of servicing can be overcome is to furnish the dealer with cars that are accurately adjusted, eliminating the necessity of service after sale and delivery are made.

With such an instrument not only can a standard of carbureter adjustment be set but it can be maintained in spite of the poor help that is often the only help available. Nearly everyone in the automobile business thinks that he can adjust a carbureter accurately. This is not a fact, from a standpoint of efficiency and economy. But even inexperienced help can adjust a carbureter accurately by using a flowmeter, as has been proved. It has proved that some old-time theories are wrong. For instance, the general opinion was that 20 to 25 m.p.h. was the best speed to drive any car economically, but this instrument proves that the speed may vary from 12 to 30 m.p.h., depending upon the make of the car and its condition.

The procedure to follow, when adjusting and servicing motor cars, is first to jack the rear wheels off the ground. Start the engine, shift into high gear, speed up to about 20 m.p.h. and then move or change the carbureter adjustment until the maximum speed is indicated on the speedometer for the least amount of gasoline that is passing through the flowmeter. Then enrich the mixture by carbureter adjustment until the speedometer shows a loss in speed. Then watching the flowmeter reading only, set the flow at one-half way between "too rich" and "too lean" by altering the adjustment. This will result in the maximum amount of economy and power, no guesswork having been employed to set the carbureter accurately. After the carbureter has been set and while the car is still up on jacks and in high gear, speed up the engine until the flowmeter indicates 1 gal. per hr. Then note the speedometer reading. Change the spark-timing until the greatest speed combined with a smooth-running engine is attained. This should be noted as being a certain number of miles per hour to a certain number of gallons per hour. If any other adjustments are necessary, they should be cared for at this time. The result will be noted in the gain or loss of speed.

To complete the final adjustment of the entire car, the work should be continued by checking up the alignment, removing the friction from the driving members and releasing the brakes so that they do not drag. The proper quantity and kind of grease should be provided for the universal-joints, the differential and the bearings and, upon completing and making the necessary adjustments, the car should again be shifted into high gear, the engine speeded up until the flowmeter registers 1 gal. per hr. and the speedometer reading noted to see if any improvement in efficiency has been achieved from the work that has been done. If all this work has been thoroughly completed, the engine and the driving members will have been adjusted to their highest degree of efficiency, and the result in miles per gallon can be considered the standard to which other cars of the same make could, and should, be adjusted when upon jacks. The car should then be removed from the jacks and driven on the road. The miles per gallon should be noted and the oil, bearings, tires and other sources of friction cared for, to eliminate as much friction as possible and the attempt should be made to obtain more speed for the same amount of gasoline. The result in the end will be the maximum number of miles per gallon to which any motor car of this make can be adjusted, and this will be the standard in miles per gallon at which all cars of this make should operate on the road. The difference between the results on the road and on jacks might be termed "road resistance." This loss in miles per gallon might be termed the "standard loss" to be expected in this particular make of automobile. We have found in this way that it costs no more to condition new automobiles up to a standard, and costs much less for service, because of being able to locate the trouble in less time. Oftentimes only one adjustment is all that is required to bring a car back to its standard number of miles per gallon.

MR. BERRY:—No matter what we think about the value of the flowmeter for the production car, it is unquestionably a fine instrument for the service manager. I have driven with a flowmeter for about a month. I find that it tells why we do not get better mileage in winter. It takes as much gasoline to start the car on cold mornings as to maintain it at boulevard speed after it is warmed up. I do not know whether it will be possible to increase very materially the number of miles per gallon in winter on cars driven in the city, because one cannot cut down the gasoline to a close adjustment in winter; however, most cars are used in summer and on long drives. For those conditions we ought to be able to get almost the theoretical maximum output. It is possible with the American carbureter to cut down so that the maximum mileage is obtained in touring with a warm engine, still having plenty of power when needed.

GEORGE A. BREEZE:—Were the performances in France rated in United States or British Imperial gallons?

MR. BERRY:—I was quoting figures that had been reduced to United States gallons. They used benzol, which gives something like a 14-per cent better mileage per gallon than gasoline.

MR. BREEZE:—We equipped a Ford car with a dash adjustment such as you mentioned. It takes as much gasoline to warm-up the engine and get the car out of the garage as it does to run 1 mile afterward. It requires 1¼ turns of opening of the fuel-adjusting valve for the first ½ mile and 1 turn for the next 1½ miles. Up to that point the average is about 16 miles per gal. Thereafter it is 25 miles per gal., with about a ⅞ turn. That is with standard equipment. Running with special equipment and a ⅞ to ¾-turn will give 31 to 32 miles per gal, and a ¾ to ⅝-turn gives 37 miles per gal. I have found that one can cut the warming-up time in half by heating the mixture. Heating the mixture has more to do with warming-up the engine quickly than anything else on Ford cars.

MR. BERRY:—I know of one foreman at the Ford factory who drove from Detroit to Indianapolis and return with his family and averaged 33 miles per gal.

A MEMBER:—Mr. Berry said that the pumping loss is one of the greatest losses at low speed. Why cannot the carbureter be developed to obviate that pumping loss by reducing the engine speed and leaning the mixture rather than by increasing the manifold suction. We have made many tests in running engines on gas instead of gasoline. To get the best economy, instead of running up or throttling the engine down, we kept leaning the mixture. In running the engines idle on gas instead of gasoline, we found that we used only about one-third of the fuel that we used when running the engines on any carbureter I have ever tested.

MR. BERRY:—The explodable range of a mixture of gas and air seems to be considerably wider than with gasoline vapor and air. The ideal carbureter I have endeavored to describe would be arranged to give the richness of mixture corresponding to the highest efficiency during the touring range; that is as far as one can go in that direction. If the amount of gasoline is decreased below that point, we not only do not increase the efficiency but we decrease it. It is impossible then to continue to lean the mixture and control the engine in that way more than the carbureters are now doing.

MR. BREEZE:—Have you ever tried stratification of mixtures?

MR. BERRY:—No.

MR. BREEZE:—I have seen some experiments in that regard in certain classes of engine.

MR. BERRY:—I know that some experiments have been made on a design that is intended to handle the mixture in that way. I believe one particular engine has two combustion-chambers. The upper one is used for idling and low-speed running, and the main combustion-chamber is for higher power. The small combustion-chamber has always been fired first, the resulting high temperature and pressure making it possible to burn very poor mixtures in the main cylinder. The compression can be kept more nearly constant in an engine of this type. This engine is only in the experimental stage and I cannot give very much information concerning it.

VAPORIZATION OF MOTOR-FUELS[1]

By P S Tice[2]

The author gives a brief and purely qualitative treatment of what a vapor is, where it comes from and how it appears; the necessity of vaporizing a liquid fuel before attempting to burn it; the separate effects of the conditions that control vaporization; and the heat-balance of vaporization. This is done to summarize the conditions surrounding and controlling fuel vaporization in the cycle of operation of a throttle-controlled internal-combustion engine, fitted with an intake-manifold and a carbureter. Charts and photographs are included and commented upon, descriptions being given of actual demonstrations that were made at the time the paper was presented. The conclusion is reached that it is well to depend as little as possible upon the cylinder heat and temperature to complete the vaporization of the fuel.

The purpose of this paper is to summarize the conditions surrounding and controlling fuel vaporization in the cycle of operation of a throttle-controlled internal-combustion engine, fitted with an intake-manifold and a carbureter. The procedure by which it is hoped to arrive at a reasonably clear and comprehensive picture of the relative values and relations of these conditions will include: (*a*) a description of what a vapor is, where it comes from and how it appears; (*b*) a demonstration of the necessity of vaporizing a liquid fuel before attempting to burn it; (*c*) a statement, with descriptions of actual demonstrations, of the separate effects of the conditions that control vaporization; and (*d*) a discussion and description of actual demonstrations of the heat balance of vaporization. These items will be given only brief and purely qualitative treatment. Generalizations dealing with the effects of impure or mixed fuels will be introduced and an attempt made to show the advantages and limitations of an engine's intake system as an environment in which to stage the vaporization process.

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[1] Mid-West Section paper.

[2] M.S.A.E.—Engineer directing the carbureter division, Stewart-Warner Speedometer Corporation, Chicago.

VAPOR AND VAPORIZATION

A vapor has been defined as a gaseous or elastic fluid phase of a volatile liquid at or near its condensation point, or below its critical point so that it can be liquefied by pressure alone; evaporation, as the change by which a substance is converted from the liquid state into, and carried off in, the vapor state; and a volatile liquid, as one that evolves vapor rapidly at ordinary temperatures.

The two terms evaporation and vaporization are commonly used interchangeably, and no sharp distinction can be drawn between them. However, it is convenient to consider the formation of vapor under natural conditions as evaporation, and vaporization as the act or process of forming a vapor by subjecting a liquid to artificially modified conditions that hasten its evaporation. From the molecular standpoint, vaporization or evaporation means the flying off of molecules against the forces of molecular attraction at the surface of the liquid, the molecules losing kinetic energy and gaining potential energy as they leave the liquid surface. The liquid and the vapor of any substance have identical compositions, and differ only in state. This is a short way of saying that the only difference is that of molecular aggregation or closeness of grouping of the molecules.

A simple demonstration with a rudimentary flash-point apparatus will serve to show that the talk of the carbureter people about the virtues of vaporization is not wholly unwarranted. To demonstrate this let us consider four tubes in which have been placed (*a*) gasoline; (*b*) gasoline from which the most volatile constituent has been removed; (*c*) kerosene; and (*d*), nothing; in other words, the last tube is a barometer. The differences between the height of mercury in the barometer and in each of the other tubes containing fuel measure the relative volatilities of the fuels. Samples of each of the three fuels, cooled to about 0 deg. fahr., are placed in three small covered beakers in the same order as in the tubes and sparks are caused to pass continually over their surfaces until combustion starts. This will occur as soon as the fuels have attained temperatures at which vapor is given off at a rate to support combustion, and the three fuels will start to burn in the order of their volatility.

Such a demonstration suggests four very important things: (*a*) the striking differences in volatility that obtain among liquid fuels; (*b*) that fuels will not burn in the liquid state; (*c*) that vapor must be present in quantities to support combustion; and (*d*) that the greater the proportion of the vapor to the liquid in a given space, the more rapid the combustion will be, provided there is sufficient oxygen present to support it.

Conditions Controlling Vaporization

It is a matter of common experience that evaporation is retarded and finally ceases if the vapor is confined at the liquid surface. This is shown in each of the barometer tubes containing only a liquid and its vapor, and in which the heights of the mercury have been constant for some time. It then becomes evident that the evolution of vapor has stopped. The reason that evaporation stops in such a case is that some of the molecules of the vapor, in their normal motions, strike the surface and join the liquid again; and, as the number of vapor molecules in a given space increases, due to the increased density of the vapor, the number of molecules so returning to the liquid in a given time will likewise increase, until finally the average number returning to the liquid will equal the average number leaving it. Under this condition the vapor is in equilibrium with the liquid.

When this equilibrium condition is reached, the space above the liquid is said to be saturated with vapor; and the density, and therefore the pressure, of the vapor are then the maxima that can exist in the presence of the liquid at the temperature of the experiment. This maximum pressure is called the saturation pressure. Proof that this pressure is the maximum that can exist is afforded by shifting the leveling bulb that forms a part of the demonstrating apparatus. The only result will be to change the vapor volume. The pressure remains constant as shown by the fixed difference in height of the mercury columns. The value of this pressure depends only upon the temperature; that is, upon the average molecular velocity of the liquid, and is unaffected by the presence of any non-combining gas or vapor. This is strictly true only for single liquids such as water or benzol. With mixed liquids such as gasoline the pressure or the temperature changes somewhat with changes in the volume of the vapor space.

Evaporation is accelerated by passing a current of air over the surface of a liquid, or by any expedient that removes the vapor from the liquid surface. This is demonstrated by the high rate of evaporation when a small stream of air is used to sweep the vapor away from the liquid surface. Since the vapor is not allowed to accumulate, it must remain unsaturated, equilibrium cannot be reached, and the liquid disappears by evaporation.

The vapor-pressure, a term denoting the saturation-pressure when used without qualification, invariably increases rapidly with a rise of temperature. A rise of pressure occurs as the temperature of the liquid is raised gently by heat from a small resistance-coil wound around the tube in which the liquid is contained. A rise of temperature is another way of saying that the molecules are moving faster; the faster the liquid molecules move the more numerous are those that fly off from the surface. To keep the liquid in the tube where heat can be applied directly to it, the leveling bulb must be raised. The change in pressure is measured by the height through which the bulb is raised.

If the temperature of a liquid is raised sufficiently to cause its vapor-pressure to become equal to the external pressure, vapor bubbles form freely in the interior of the mass of liquid, by the well-known process of boiling. The temperature at which this occurs, under standard atmospheric pressure, is called the boiling-point.

When a flask contains fairly hot water and is in open communication with the atmosphere, the water is at a high enough temperature to exert a vapor-pressure nearly equal to that of the atmosphere. A further rise of temperature, which can be brought about by setting the flask on a hot plate, results in the formation of vapor bubbles as soon as the pressure of the water vapor equals that of the atmosphere.

If the external pressure remains constant, the temperature of a pure boiling liquid will remain constant at its boiling-point, while the liquid passes off by vaporization. When water at sea level boils vigorously, a thermometer inserted in it will continue to register 212 deg. fahr. as long as the boiling continues. It is evident that water in the vapor state is leaving the surface and does not return to it. But, if the liquid is contained in a closed space, it can be made to boil at temperatures below its normal boiling-point by reducing the pressure. This is made evident when the liquid in the vapor tube is allowed to cool slightly below its normal boiling-point. If the pressure in the tube is reduced, as is the case when the leveling bulb is lowered, the liquid will boil again at its now reduced temperature, as indicated by the bubbles seen to form in it. The same phenomenon can be shown by removing from the hot plate the flask in which water is boiling, sealing it and applying ice to the vapor space above the liquid. In this case the ebullition will be more spectacular than in a tube, because the mass of liquid is greater and the pressure can be reduced conveniently to a much lower value. As the boiling progresses, the temperature will drop rapidly. Removal of the ice, the presence of which causes the pressure reduction, will stop the boiling, while the replacement of the ice will cause the boiling to start again. The boiling may be made to continue until the temperature of the liquid is about 20 deg. fahr. below that of the room.

Correspondingly, if the pressure is raised, the temperature of a liquid must be raised above its normal boiling-point to secure ebullition. There is for every temperature a corresponding equilibrium or saturation-pressure of the vapor, and vice versa. But the temperature and pressure cannot be raised indefinitely. As the temperature is raised, the density of the saturated vapor increases, while that of the liquid decreases. If the temperature is raised sufficiently, the densities of the liquid and of its vapor become equal at a definite temperature depending upon the substance. This temperature is called the critical temperature, because it is the limiting temperature at which a separation of the liquid and the vapor states can be observed. If we seal a liquid and its vapor in a small tube and raise the temperature of the tube, a point will be reached at which it will be impossible to distinguish between the liquid and the vapor. Reducing the density of the liquid causes its upper boundary to rise until the liquid completely fills the tube. All parts of the tube are now shared equally by the liquid and its vapor. When the temperature drops, the tube, from bottom to top, becomes filled with a fog of reaggregated molecules, and the meniscus separating the two states rapidly settles as the density of the liquid increases.

At temperatures above the critical temperature there is what is known as continuity of state, since it is impossible to cause separation of the two states by modification of the pressure alone. Attention is directed to Fig. 1, which is a plot of isotherms on pressure-volume coordinates for carbon dioxide. The properties shown in this graph are characteristic of all substances. It is a plot of the typical pressure-volume relations of a substance at temperatures below and above the critical value. The horizontal portions of the isotherms enclosed by the dotted curve show the characteristic change of state, from liquid to vapor, or vice versa, at constant pressure and constant temperature. To the right of the horizontal portions, the substance is completely vaporized and behaves like a gas. To the left, it is all liquid. Note that as the critical temperature is approached there is a continually smaller change of volume between the all-liquid and all-vapor conditions. This illustrates a statement to be made later to the effect that the latent heat becomes less with increased temperature, and at the critical temperature has a zero value.

The two controlling factors in the process of vaporization are the temperature of the liquid and the vapor-pressure exerted on its surface. Raising the temperature at a constant vapor-pressure, or lowering the vapor-pressure at a constant temperature, or doing both simultaneously, causes a further evolution of vapor from the liquid. This is the sole means available for accelerating the vaporization of a liquid.

The controlling pressure in vaporization is only the pressure exerted by the vapor upon the liquid surface. A demonstration of the fact that the saturation pressure of a vapor is unaffected by the presence of any non-combining gas or vapor makes clear what is meant by the term partial pressure. This can be done by using a normal barometer tube devoid of air above the mercury. If a small quantity of liquid is introduced into this tube, its vapor at once fills the space above the mercury, and the mercury column falls an amount proportional to the vapor-pressure of that liquid at the prevailing temperature. When a vapor-pressure tube from which all air has been expelled, is used as a barometer, if the tube cock is opened, an amount of air admitted, the tube resealed, and the bulb returned to its former position at which the barometric height was shown, the pressure above the mercury will be higher than before. The difference between the new height of the mercury column and the barometric height is a measure of the pressure due to the air in the tube. If, now, we introduce into this tube some of the same liquid that was put into the conventional barometer tube, and set the leveling bulb so that the volume above the mercury in the tube is the same as before the liquid was admitted to it, it will be found that the

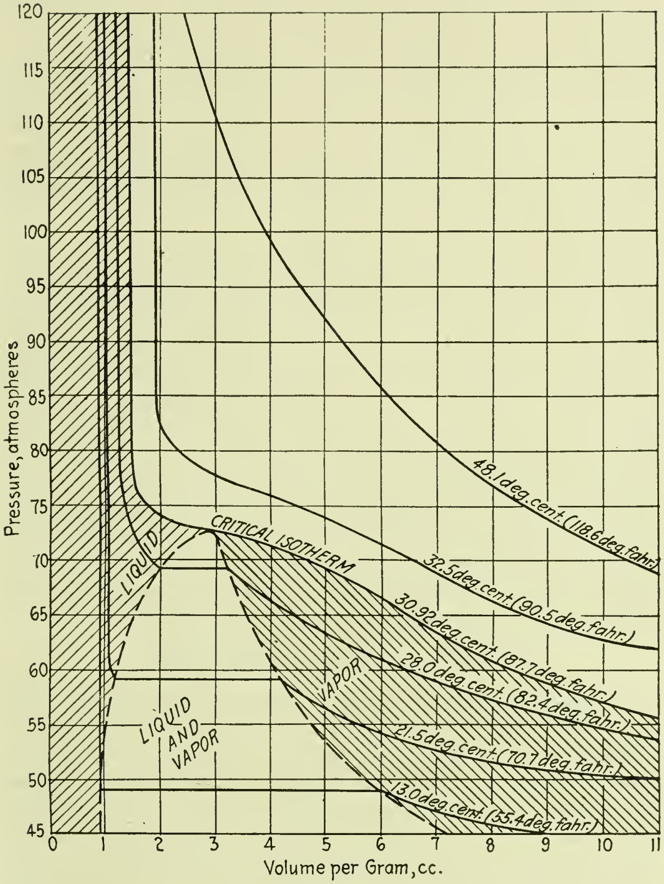

Fig. 1—Series of Curves Showing the Typical Pressure-Volume Relations of a Substance at Temperatures Above and Below the Critical Value

pressure-rise due to the vapor in the tube containing air is exactly the same as that in the first barometer tube which contained no air when the liquid was put in.

This shows that a given equilibrium-pressure exists for a given liquid at a given temperature, whether that vapor-pressure is the total pressure exerted on the liquid surface, or it is only a part of the total pressure so exerted. The only difference to be noted in the two cases is that the equilibrium pressure is attained much more rapidly when only the vapor occupies the space above the liquid. The molecules of a gas interfere with the free dispersion of the vapor molecules, causing a greater average vapor density at the surface of the liquid in one case than in the other.

When a vapor is in equilibrium with its liquid and the vapor exerts only a fraction of the total pressure, a change of pressure of the total fluid above the liquid, such as is produced by a change of volume, will change the pressure of the vapor in a like ratio and destroy the equilibrium relation. But, if the temperature of the liquid is kept constant, equilibrium will become reestablished at exactly the same vapor-pressure as before. Depending upon whether the volume is increased or decreased to change the total pressure, the partial pressure of the vapor, and therefore the relative vapor-content of the space, will become correspondingly greater or less. This is a matter of great importance in the practical carburetion of an engine, as will be shown later.

HEAT BALANCE OF VAPORIZATION

The conception of heat as a form of energy, the common manifestation of which is temperature, is familiar to everyone. Degrees of temperature represent intensities of heat and not quantities of heat, because, when equal quantities of heat are imparted to two bodies of equal mass but of unlike substance, one is found to be hotter than the other. The specific heats of the two substances are then said to be different, the one showing the higher temperature having the smaller specific-heat capacity. Besides raising the temperature, heat usually causes an increase of volume. Part or the whole may go toward producing changes of state by fusion or vaporization and may cause chemical reactions. Part may also be transformed into other forms of energy, producing the phenomena of light, electricity and the like. Heat may be imparted to a body (*a*) by conduction, as along a metal rod; (*b*) by convection, as through the rooms of a house by air currents; and (*c*) by radiation, as from the sun to the earth.

To convert a substance from a liquid to a vapor without change of temperature, it is necessary to impart to it a certain amount of heat. The quantity of heat so required per unit mass of the liquid is the latent heat of vaporization. It is called latent because it causes no temperature-rise and disappears, so far as our senses are concerned, in performing the change of state. Since vaporization of a liquid produces a great increase in volume, which acts against a definite pressure, external work must be done by the vapor as it is formed. In the performance of this work the heat disappears or becomes latent, as was demonstrated by the water in the flask set on a hot plate boiling without any change of temperature. The latent heat of vaporization diminishes with increasing temperature, and becomes zero at the critical temperature, where the distinction between a liquid and a vapor vanishes.

The total heat of vaporization of a saturated vapor at any temperature is the quantity of heat required to raise a unit mass of the liquid from any convenient zero to the temperature in question, and then to evaporate it at that temperature under the constant corresponding pressure of saturation. If, instead of converting a liquid into its vapor, we reverse the process or condense the vapor into its liquid, and bring the temperature of the recovered liquid back to the value from which it was initially raised, the total heat of vaporization is recovered and transmitted from the liquid to some other body.

Let us take two bulbs, sealed to each other in the form of an inverted U, and containing only water and its vapor. The water is all in the outside bulb, and the other one, immersed in the beaker, contains only water vapor. Applying heat to the exposed bulb causes the water to vaporize, making heat latent in so doing. The vapor thus evolved is condensed in the other bulb, in which liquid begins to appear. Upon condensation of the vapor, the heat that was latent in it reappears and is taken up by the walls of the bulb and the air in the beaker surrounding it. That heat will reappear at the bulb immersed in the beaker can be made evident from observation of the convection currents set up in a liquid poured into the beaker and the vapor coming off that liquid. With suitable precautions to prevent loss of heat to other bodies than the liquid in the beaker, the whole heat made latent in the vapor formed in the outer bulb would be recovered at the immersed one.

If a liquid is vaporized by the rapid removal of its vapor, without applying heat at a rate equal to that at which the change of state makes it latent, the temperature of the liquid falls rapidly, and it takes what heat it can from its surroundings. To demonstrate this we can place a small tube containing water inside a larger one containing a small amount of ether. Blowing air through the ether in the outer tube removes its vapor rapidly, and the ether evaporates and cools. Continuing the process for a moment so lowers the temperatures of both liquids that the water freezes. This drop in temperature is explained on the ground that the liquid molecules, which are moving most rapidly, are the first to fly off from the surface. Hence, the average kinetic energy of the molecules left behind is less than the initial average for the liquid. Since the temperature is a function of the average molecular velocity, the liquid is cooled by the evaporation. In the cases of liquids having high latent-heats of vaporization, it is possible by this means to lower the temperature of the vaporizing liquid so far that it will solidify or freeze long before its vaporization is complete.

Again using a sealed double bulb containing only water and its vapor, if the water is all run into one of the bulbs and the other is plunged into a freezing mixture, the vapor will be removed so rapidly from the water surface by condensation in the cold bulb, thereby reducing the vapor-pressure throughout, that the temperature of the water in the exposed bulb will drop to the freezing point and the water will solidify into a cake of ice. This apparatus is called a cryophorus, from the fact that when it is treated in this manner it bears frost.

The points it is desired to make are: (*a*) that vaporization is accomplished only as a result of the application of heat; and (*b*) that the vaporization of a given quantity of a given liquid requires a definite amount of heat, no matter how the other conditions by which the process is surrounded may be altered.

MOTOR-FUELS ARE HETEROGENEOUS MIXTURES

All present-day motor-fuels are mixtures of a great number of substances. Each component has, of course, its own particular heat of vaporization and vapor-

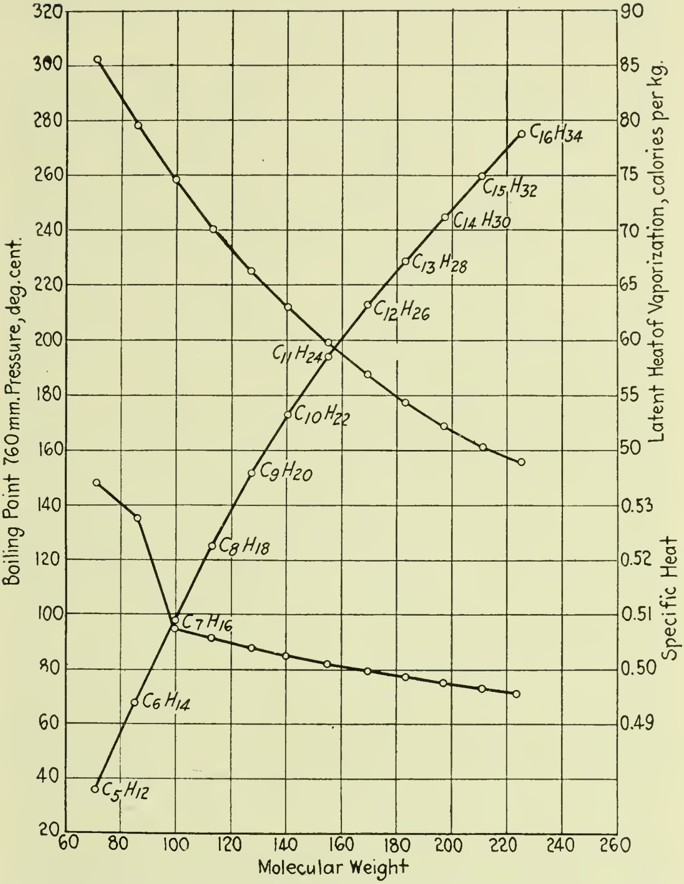

FIG. 2—GROUPING OF PHYSICAL CONSTANTS REFERRED TO MOLECULAR WEIGHT FOR THE MEMBERS OF THE PARAFFIN SERIES FOUND IN OUR FUELS

pressure. Fortunately for both producer and consumer, the vapor-pressures of these mixtures agree very closely with that of the most volatile constituent. The result is that, if we pump enough fuel into an engine, we can get it started on the trace of volatile material in the fuel. Attention is directed to the result with the two kinds of gasoline in the first demonstration.

The specific heats do not vary greatly among fuels or their constituents. Among petroleum fuels latent-heat values decrease with decreasing volatility. Naturally, the less volatile the fuel components are, the higher will be the temperature at which a given relative vapor-density is attained. Assuming that our interest is in the maintenance of a given rate of vaporization of the fuel, this fact entirely overshadows the advantage that might be expected from a consideration of the latent heats alone. Not only is a greater total heat needed to vaporize a less volatile fuel but less heat is made latent in the process. As the fuel becomes less volatile, the normal intake temperature in an engine will of necessity increase if a given relative vapor-content is to be maintained in the charge. The relations of the pertinent physical constants of the majority of the paraffin petroleum compounds appearing in motor fuels are shown in detail in Figs. 2 and 3.

Fig. 2 is a grouping of physical constants referred to molecular weight, for the members of the paraffin series found in our fuels. These values have been taken from the work of C. F. Mabery and A. H. Goldstein.[3] Fig. 3 shows an interesting heat relation developed from the values shown graphically in Fig. 2. The facts here presented are: (*a*) the great increase in total heat of vaporization in the face of lowered values of the latent heat accompanying reduced volatility; and (*b*) the enormous increase in the sensible heat expressed as a fraction of the total heat, as the volatility is reduced. When it is considered that within the last 5 or 6 years the mean characteristics of our motor gasoline have shifted from those of hexane C_6H_{14} to somewhere in the immediate neighborhood of those of octane C_8H_{18}, the reasons for our need to apply more heat and to maintain higher charge temperatures are explained somewhat by this graph.

VAPORIZATION IN THE ENGINE INTAKE

To apply the facts that have appeared thus far, and to relate them to the conditions in the intake system of an

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[3] See *American Chemical Journal*, vol. 28, pp. 66 and 165.

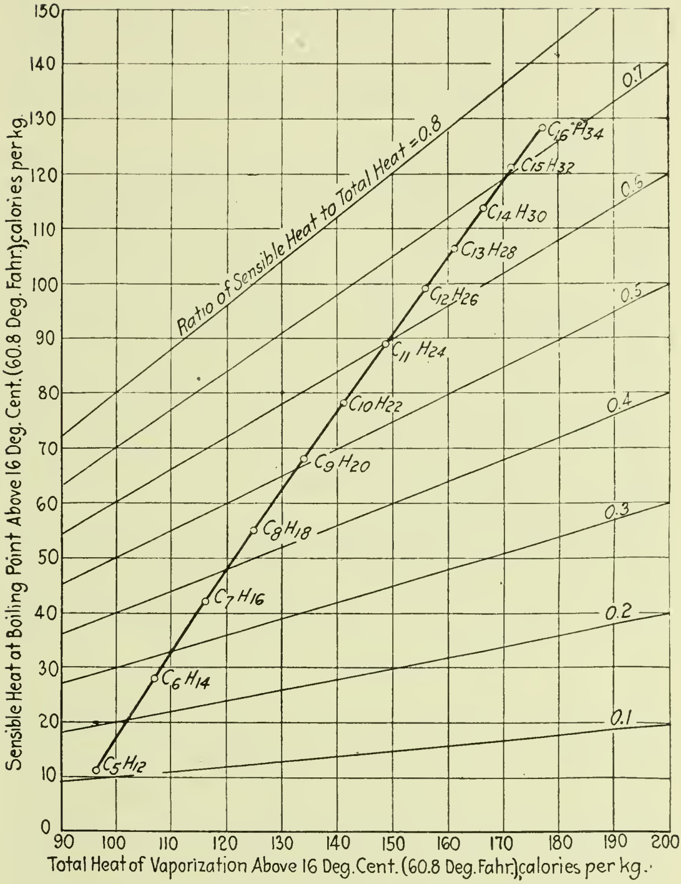

Fig. 3—Relation of the Total Heat of Vaporization to the Sensible Heat

engine, it is necessary only to examine those peculiarities of the intake that modify the rate of evolution of vapor. The simplest way to do this is to follow the fuel from the carbureter nozzle to the cylinders. As soon as the fuel issues into the airstream, some of its molecules fly off into the gas space, in numbers depending upon the temperature of the fuel. Since the partial pressure due to the fuel vapor is zero, or at least is negligibly small, the number of molecules leaving a unit surface in a unit time is no doubt the maximum possible at that temperature of the liquid.

Let us imagine two fuel-sprayers passing fuel at the same rate and temperature, the only difference between them being that one causes twice the initial liquid-surface exposure as the other. It is evident that twice the number of molecules will fly off into the gas space in one case as in the other. A very important expedient for accelerating the rate of vaporization is indicated here; and it can be stated that the initial rate of vaporization is directly proportional to the extent of the liquid surface exposed. But, as we have seen, heat disappears in doing the work of vaporization. Thus, from the first instant of issuance into the air-filled space, the temperature of the liquid will fall unless a suitable quantity of heat is imparted to it.

If twice the number of molecules fly off initially from the liquid discharged from one fuel sprayer as from the other, it is evident that a suitable quantity of heat to maintain the temperature of the liquid must be imparted at twice the rate in one case as in the other. The quantity of heat used in vaporizing a given amount of liquid fuel is the same in both cases; but, to maintain constancy of temperature, the rate of heat supply must be in direct ratio to the area of the liquid surface exposed. If heat is not supplied to maintain the temperature, it follows that the initial rate of heat loss from the liquid fuel will be twice as great in the case of the sprayer in which twice the surface is exposed. Since the temperature varies as the heat-content of a body, it is seen that the initial rate of temperature drop in the liquid fuel is in direct proportion to the surface exposed.

Allowing the temperature to fall in this way lowers the saturation-pressure and so causes a nearer approach to equilibrium between vapor and liquid, with a consequent reduction in rate of vaporization. Considering the discharges of the two sprayers, if no external heat is applied it is evident that equal drops in temperature mean equal amounts of fuel vaporized, and that ultimately the temperatures and the vapor-densities will be the same. But there is no time for dalliance in the intake of a modern engine; and so it is that, within the time available in the system, greater extension of the fuel surface causes a greater lowering of the temperature and also accounts for a greater vapor-density in the charge.

Let it now be supposed that the initial rate of vaporization is insufficient, and that heat is applied in excess of that needed to maintain the initial fuel-temperature. The saturation-pressures are then raised, the equilibrium condition is deferred and the rate of vaporization is accelerated in consequence. But the rate of passage of heat into a body varies directly with its exposed surface; and the rate of vaporization varies directly with the rate at which heat is taken on. Thus it is seen that the sustained rate of vaporization is directly proportional to the liquid surface exposed.

In the case of an engine and intake system arranged to apply such an excess of heat that the final charge-temperature is well above the initial temperature of the fuel, as in a typical modern engine, the direct result of an appreciable extension of the liquid surface will be to increase the vapor-density of the charge, or to lower the charge-temperature, or to accomplish both these things simultaneously.

Effect of Intake Pressure

The general effect of pressure on vaporization will not require discussion at this time. However, *changes of pressure in the engine intake are of extreme importance*, particularly those following manipulation of the throttle.

Let us suppose an engine to be throttled, as in driving a car within the speed limit on a smooth level road. The manifold pressure is then approximately one-half an atmosphere. Only a small amount of air is passing to the engine. Under these conditions the relative vaporization rate will be substantially at its maximum, since the pumping strokes of the engine withdraw the vapor from the liquid surface as fast as it is formed, and maintain a low total pressure of which the vapor-pressure is only a part. The result is that relatively little liquid exists in the manifold, and its walls are as nearly dry as they ever can be.

Now let us open the throttle somewhat. The immediate results are rise of pressure and the passage of more air and more fuel. From a vaporization standpoint, the circumstances are: (*a*) increased density of the fuel vapor at the liquid surface; (*b*) reduced relative surface exposure of the liquid; and (*c*) lowered intake temperature. Each of these items is a powerful deterrent to vaporization; and the net result of opening the throttle is that the relative vapor-content of the charge, upon which we depend for regular ignition and good performance, will be much reduced until such time as the intake walls have become more extensively wetted, increasing the exposed surface of the liquid fuel and thereby partly restoring the vaporization rate. In the conventional intake system, the throttle-valve is interposed between the intake-manifold and the sprayer or nozzle of the carbureter. As the fuel passes the nearly closed throttle it is subjected to extensive spraying, and therefore to extension of its surface, which spraying is by no means equalled by that at the ordinary nozzle under any condition of operation. Attention is directed to Fig. 4 which gives a very good idea of what happens at the throttle of a carbureter at its several positions. This photograph was published in my paper on Intake Flow in Manifolds and Cylinders.[4] Attention is called to the marked differences in agitation and churning-up of the fuel as the throttle position is changed.

If the throttle is now closed to bring us again within the speed limit we accomplish: (*a*) reduction of density of vapor at the liquid surface; (*b*) extension of the exposed liquid surface; and (*c*) rise of intake temperature. Each of these is a powerful accelerator of vaporization; and, opposed to them, the heavy accumulation of liquid on the manifold walls does not stay there very long, but goes into the cylinders and causes excessive enrichment.

VAPORIZATION IN THE CYLINDERS

The question as to what fraction of the fuel supply can enter the cylinders as liquid and still be completely burned is almost impossible to answer. No doubt the size of the admissible unvaporized fraction varies chiefly with the fuel, but it varies also with the extent of its mechanical division, with the temperatures of the walls it falls upon, with the temperatures to which it is raised and with the pressures to which it is subjected during compression and the early part of the combustion.

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[4] See TRANSACTIONS, vol. 16, part 1, p. 397.

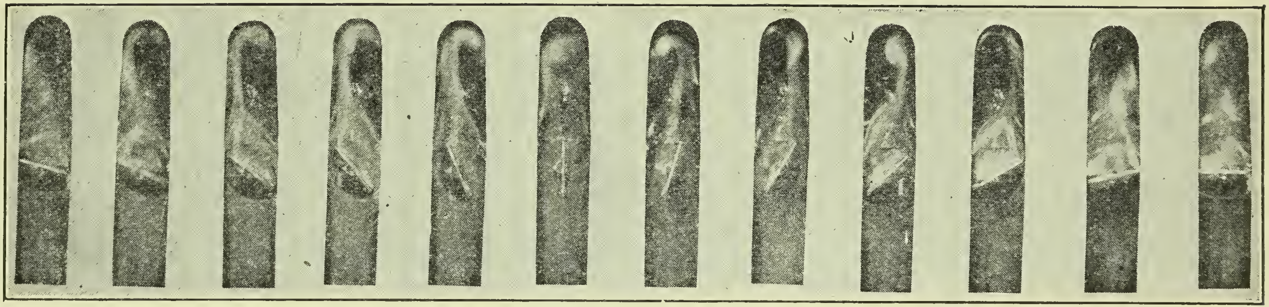

FIG. 4—SERIES OF PHOTOGRAPHS SHOWING THE ACTION THAT TAKES PLACE IN THE INTAKE-MANIFOLD AT AND ABOVE THE THROTTLE AS ITS POSITION IS CHANGED FROM CLOSED TO FULL OPEN AND BACK TO CLOSED

The term polymerization is defined as the act or process of changing one substance into another which has the same composition but a different molecular weight. The polymerization products of petroleum compounds attain extraordinary molecular weights; the very heavy compounds are resinous and tarry, and their combustion is almost impossible. The increased rate of carbon-deposit formation that has been generally experienced during the past year is usually explained on the ground of high polymerization of our later fuels.

If the fuel is one that does not break-down, and does not polymerize, it is entirely possible that the liquid surviving the compression will vaporize or may attain its critical temperature during the combustion; and will then burn rather well along in the expansion stroke, appearing on the card as a sustained combustion, or even an after-burn. Most cards from automobile engines, when plotted on logarithmic pressure-volume coordinates, show an appreciable length of the early expansion-line to have a slope close to unity, as in Fig. 5. This can mean only that the combustion is continuing at a rate sufficient to maintain the temperature of the gases at a nearly constant value, in spite of the expansion.

Fig. 5 shows a typical part-throttle indicator-card plotted on logarithmic coordinates. The data from which this plotting was made were obtained at the Bureau of Standards, with the point-to-point diaphragm-type indicator developed at the Bureau under the direction of Dr. H. C. Dickinson. The graph gives a good idea of the closeness of the work that can be done with this form of indicator. In such a plot as this one, the exponent in the expression for adiabatic compression or expansion is scaled off directly. Note the change in slope of the expansion line, to which attention was directed previously. Also note the small value of the exponent n for the compression stroke. If there were no heat losses from the compressing gases, this n would have a value of 1.41. But, of course, there are heat losses, and the major one is, no doubt, the heat loss to the fuel that is being vaporized during this part of the cycle.

With our present fuels, it is well to depend as little as possible upon the cylinder heat and temperatures to complete the vaporization of the fuel. Most of our fuels break-down too easily, and many of them polymerize extensively. Both these things cause rapid fouling of the combustion spaces, and particularly of the intake-valve

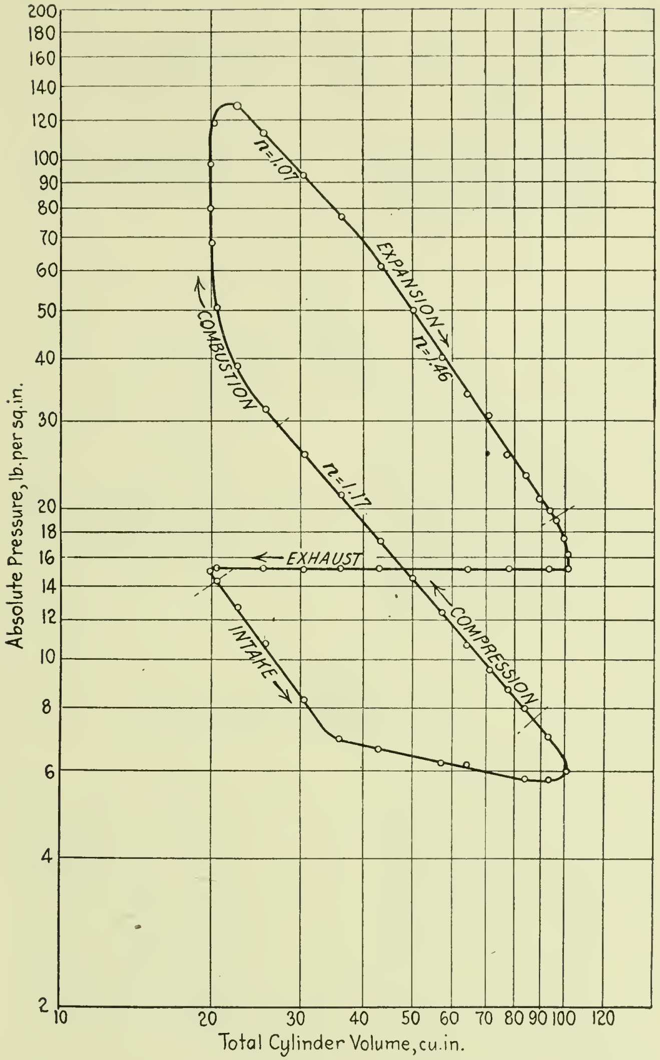

FIG. 5—TYPICAL PART-THROTTLE INDICATOR-CARD PLOTTED ON LOGARITHMIC COORDINATES

heads, since the latter are thoroughly wetted by unvaporized fuel.

A further argument is that the greater the dependence upon vaporization in the cylinders, the greater will be the liquid mass in the intake-manifold to be distributed among the cylinders and, obviously, the more difficult it will be to secure its equal distribution. Small errors in the distribution of the liquid in the manifold represent very gross ones in mixture proportions within the cylinders. But, no matter how much liquid fuel we may be willing to have in the intake passages and cylinders, there always must be sufficient vapor in the charge at the time of ignition to support combustion, or the engine will not operate. Bearing in mind the valuable effects of (*a*) extension of liquid surface; (*b*) rise of liquid temperature; and (*c*) reduction of vapor-density at the liquid surface, and having considered the conditions in the intake passages,, it is interesting and useful to examine the interior of the cylinder.

In the intake, the fuel is sprayed at the carbureter nozzle, churned-up at the throttle and spread over the passage walls to increase its surface. This increases not only the area from which vapor can emanate but also the area through which heat can pass to the liquid to maintain or raise its temperature. Furthermore, at the same time the fuel is being spread out, as it were, its vapor is being swept away from it, accelerating its vaporization and increasing its ability to absorb heat by lowering its temperature.

Referring again to Fig. 4, the throttle is here taken through a complete cycle of 180 deg. Notice the differences in agitation, churning-up and, therefore, extension of liquid surface among these several throttle positions. It is perfectly obvious that the greatest spraying effect is obtained with the least throttle-opening.

By the time the liquid that is left gets into the cylinders, the density of the vapor in contact with it is greater than was the case in the intake passage. The sources of heat supply with which it now comes into contact are greater and may have higher temperatures than those of the intake. For this reason, vaporization may progress right up to and even after ignition. No doubt it does so under some conditions of operation, no matter what the fuel; and perhaps it does so under all conditions of operation with some fuels. However that may be, if the characteristics of the fuel and the temperature gradients

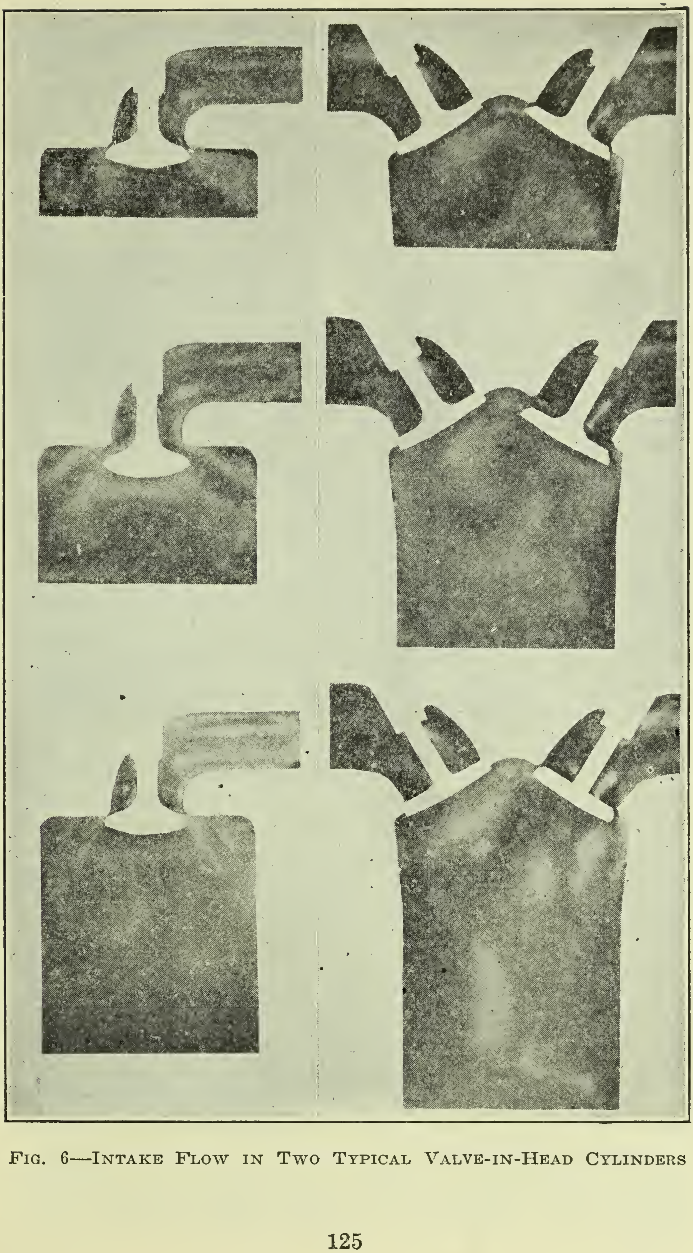

FIG. 6—INTAKE FLOW IN TWO TYPICAL VALVE-IN-HEAD CYLINDERS

and the pressures that exist permit further vaporization, the only way to carry it forward at the highest possible rate is to extend the liquid surface, just as in the intake. The greatest effect of this sort must follow the highest possible rate of homogeneous and therefore orderly internal motion of the charge. This motion is described as orderly turbulence. That the turbulence in the intake may be reasonably orderly and at the same time high, is clear.

Let us look into a few cylinders, as shown in Figs. 6 and 7. In Fig. 6, we have two valve-in-head cylinders of different design. On the left is an ordinary valve-in-head type and at the right is something approximating the Liberty cylinder. The three stages shown represent positions of the piston on the intake stroke. Note the turbulence or flow-lines in the upper views, where the intake-valve is open only a small amount and the piston is just starting down. In the next stage the intake-valve is wider open, the piston farther down. In the last stage the intake is nearly closed and the piston clear down. Note how the turbulence or internal motion fades as we approach the end of the intake stroke. This condition is even more marked in the more conventional form of cylinder that finds such an extensive application in the present-day automotive vehicles.

In Fig. 7 we have two L-head cylinders of different combustion-chamber shapes. One can follow the flow-lines in this illustration without difficulty as they are easily distinguished. Note that the change in slope, as we may call it, of the top wall of the combustion space changes the volume of the cylinder that is swept by an orderly and therefore useful internal motion. The turbulent volume is flattened out in one instance but, with the other shape, nearly the whole contents of the cylinder participate in the motion.

These views show that the turbulence is more orderly and more homogeneous, and therefore more active and useful, in some forms of cylinder than in others. But in no case does the cylinder turbulence approach that of the intake. High cylinder-turbulence is a desirable thing and should be realized to the utmost in all engines; but, because of the limitations imposed upon its usefulness by fuel characteristics and by the values and distributions of the cylinder temperatures, it cannot be greatly or consistently relied upon to build up a usable vapor-content in the charge.

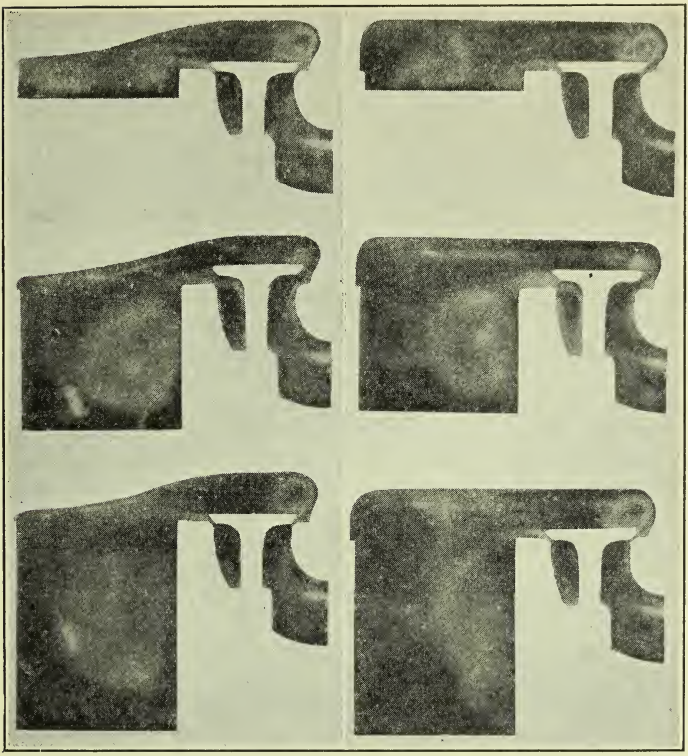

FIG. 7—INTAKE FLOW IN TWO FORMS OF L-HEAD COMBUSTION SPACE

THE DISCUSSION

CHAIRMAN H. L. HORNING:—Will Mr. Tice repeat the partial-pressure demonstration with the barometer tube?

P. S. TICE:—The demonstration is made with a tube having within it an unoccupied space; that is, it is a normal barometer. The introduction of a liquid into the tube fills the space with vapor and, because of the temperature at which the liquid exists, the vapor has a certain pressure, which is represented by a fall in a mercury column. This fall is read and noted. When the contents have been expelled and the tube sealed, the vapor-pressure tube becomes a barometer also. By keeping the leveling bulb down and using the apparatus as a barometer, air is admitted which raises the pressure. The change in the height of the mercury column represents the increase of pressure in the tube, due to the air. Pouring some of the same liquid used in the barometer tube into the vapor-pressure tube by means of a funnel and then readjusting the leveling bulb so that the air and the vapor together occupy the same volume that the air alone occupied before the introduction of the liquid, it is found that the change in pressure due to the vapor is the same both in the barometer tube having no air and in the vapor-pressure tube having air.

Chairman Horning: — If another liquid were introduced, there would be a still greater drop; and the introduction of still another liquid would cause another drop. The fundamental lesson in this is that there are the same number of molecules in 1 cu. in. of any gas at the same temperature and pressure. That is a very important law. If we continue to introduce more liquid, the pressure continues to rise because the pressure is controlled by the number of molecular bombardments there are in a unit of time, and pressure refers to how fast the molecules are going. In this demonstration, as soon as heat was introduced the velocity began to increase, and the heat absorption appeared in the velocity of the particles, which were thrown into a vapor state. When they cooled down, what the cooling down meant was that they had come into touch with something that was moving rather slowly and that this had taken the velocity out of them. But the slow-moving object also gained a little velocity, and finally it had as much velocity as the particles. The liquid state and the gaseous state were then practically the same.

Frederick Purdy:—How were the photographs taken of the phenomena inside the cylinder and the manifold?

Mr. Tice:—Before undertaking the work, we decided that the flow in both halves of a symmetrical passage, such as the intake passage, would be symmetrical when the halves were divided by a plane; and that we could put such a plane through the passage without destroying the flow-lines. In other words, flows coming through the back of the passage normal to a glass plate would be met by equal flows from the other half of the passage. We split some intake pipes down the middle and cemented them on glass plates, connected the outlets of those pipes to an engine intake, pumped air through them by motoring the engine, sprayed liquid into the air entering this passage, had throttles in place, and proceeded to photograph what we saw. The same method was followed in the cylinders. What we showed as happening in the cylinders was probably not so close to the truth as it is in the case of the intake, because of the unsymmetrical shape of the cylinder-head, particularly in the case of the L-head type.

W. F. PARISH:—In reference to the statement in Mr. Tice's paper that "The increased rate of carbon-deposit formation that has been generally experienced during the past year is usually explained on the ground of high polymerization of our later fuels," I have been making some very careful examinations of my records of lubricating oils. In assembling these data I have found a most extraordinary state of affairs in connection with present engine lubricants as compared with the same grades some years ago. Liquid fuels are becoming heavier gradually but we do not know to what extent the lubricants are becoming heavier. The grading of motor oils has remained the same, but a careful examination of the averages of the lubricants placed on the market in the past year shows that the viscosity of these oils has been growing gradually greater with the increased heaviness of the fuel. I refer to the increased public demand for heavier oils to compensate for the thinning-down of the lubricant in the engines. The use of heavier oils causes more carbon, a large part of which comes from the heavy stocks that are put in the oils to increase the viscosity.

MR. TICE:—That is true. There is evidence to show that a large part of the increase in carbon formation in the last year is attributable also to polymerization of the fuel.

CHAIRMAN HORNING:—That is caused by the increased content of unsaturated fuel, is it not?

MR. TICE:—Somewhat, but not entirely. The things do not go hand-in-hand with all kinds of petroleum; they do with some.

CHAIRMAN HORNING:—The higher the content of unsaturated fuel, the greater the polymerization is apt to be. The unsaturated fuels are the ones that have the two extra hydrogen atoms left out. Some are called olefins, but as they become heavier, they have a tendency to break in two and make a more complicated substance. They do just the opposite to cracking; the cracking breaks-down into the simpler system and the polymerization steps-up into the more complicated. Anyone who has had an engine with the intake opening before dead center will know what I mean. In such an engine the hot gases are pushed back into the intake passages and the valve will coat-up rapidly with a gummy substance, like varnish. We had an engine go wrong in that way and we found that the timing was wrong, although I tried to prove that the fuel was wrong. The deposit on an intake-valve is due to polymerization. In order to find what causes polymerization, we tested many unsaturated fuels in the laboratory and found both moisture and oxygen to be necessary. We found that an almost imperceptible amount of moisture would start the trouble; but there was no trouble when the fuel was absolutely free from moisture or any oxygen-bearing material.

MR. TICE: — This critical-temperature phenomenon is something with which most of us are unfamiliar; it should be observed in detail.

B. STOCKFLETH:—What is the time element for vaporization in the manifold?

MR. TICE:—The time available for vaporization in the intake is only the time that is required for the liquid to travel from the carbureter nozzle into the intake. It depends upon the velocity of the fluids in the intake and the length of the intake.

MR. STOCKFLETH:—If a certain amount of kerosene is put into the manifold by the carbureter nozzle, and if there is a heated chamber into which it will pass, it will take a certain time before the fuel will become a dry gas?

MR. TICE:—Yes. The time required for that to be accomplished depends upon the rate at which the temperature is raised, the extent to which the fuel is spread out, the area of the surface and the rate at which the vapor is taken away from the liquid. The more useful way to consider the rise of temperature of the liquid is to look at it from the point of view of the rate at which we put heat into it, although the controlling factor is actually the temperature of the liquid.

MR. STOCKFLETH:—In an experiment I am making at present, instead of using the ordinary carbureter-nozzle that creates a fog in the manifold, I poured the kerosene through a fixed orifice into various channels or troughs and let it vaporize only as the heat came to it from the exhaust gases surrounding these pockets. In that way I was able to get a very much clearer mixture than I did by having it pass through the carbureter. This was evident when looking through a peep-hole.

MR. TICE:—Yes, there would be a dry mixture under those conditions. We would all like to do it that way, but can you do that at a useful rate?

MR. STOCKFLETH:—I kept the fuel at a constant flow. I was not attempting any great degree of flexibility, but was surprised by the flexibility shown.

MR. TICE:—I mean, could you run an engine by this method and accelerate it? Could you do the necessary things?

MR. STOCKFLETH:—The proposition is only experimental. It shows that the carbureter action is not needed except for flexibility.

MR. TICE:—Exactly. You accomplished the result by doing exactly the same things as in a conventional arrangement, but gave them a little different order of prominence. You cut out practically the extension of surface and did it entirely by sweeping away the vapor and by heating the liquid. Of course, the rate at which you evaporated the fuel was much lower and the weight of fuel evaporated per unit of time was much less than if you had sprayed the fuel.

J. W. STACK:—Has Mr. Tice determined within a reasonable range what vapor-tension is necessary in a fuel in order to have sufficient vapor to turn over an engine at zero atmospheric temperature?

MR. TICE:—I have never attempted to measure that. I do not see what good it would do us to know it. We are limited to certain sets of conditions.

MR. STACK:—In other words, gasoline can be made with a certain percentage of hydrocarbons boiling over below a certain temperature. What is the maximum or the minimum percentage of those light boiling-points necessary to produce combustion to heat the engine at a low temperature?

MR. TICE:—I do not know. I have never attempted to determine that. So long as the fuel has components that will flash at our starting temperatures, we can start by choking enough fuel into the engine. In his booklet[5] on Economic Utilization of Fuel, Prof. C. A. Norman has computed the mixture temperatures at which we can support as vapor the necessary fuel to burn, for fuels of different mixtures.

As stated in his paper on the Condensation Temperatures of Gasoline and Kerosene-Air Mixtures,[6] Prof. R. E. Wilson has made experiments and obtained some data on the temperatures at which different ratios of fuel mixtures can exist and still have all the different fuels in the vapor state. These data are all interesting and useful but they do not tell us how to arrive at that condition. They tell us how low the temperature may go and the fuel still be in condition to burn, but they do not tell us how to reach that condition within the time available. The rate of vaporization is our limiting factor. Professor Wilson emphasizes this point. What we need is a higher rate of vaporization. Professor Norman discusses the desirable height of mixture temperatures, the results of using higher temperatures, the losses to be sustained and the like. If we could devise some compact apparatus that would evolve vapor at a rate to permit us to form the charge and keep it dry, we could work at the temperatures computed or determined experimentally; but, because we cannot usually do that in practice, we necessarily must work at a higher temperature. The interchange of heat ordinarily is not complete by the time we have to burn the liquid. Mr. Stockfleth's experiment is right in that respect. He obtained a drier charge by allowing more time. He took much longer to evaporate the liquid. If we apply such an apparatus as that to an engine, it must be an apparatus involving a considerable time-interval.

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[5] See Ohio State University Bulletin, No. 19.

[6] See THE JOURNAL, November 1921, pp. 313, 314 and 318.

Mr. Stockfleth:—In regard to Professor Wilson's and Professor Norman's determinations of the condensation points of kerosene and of ordinary gasoline, as shown by tests performed recently, I find, by looking through the peep-hole, that even at a temperature of about 110 deg. fahr. above the condensation temperature as given by Professor Wilson, a slight fog is indicated in the manifold. The temperature Professor Wilson gives as the condensation-point for a dry mixture of kerosene is about 220 to 230 deg. fahr. The maximum mixture temperature in my experiment was 195 deg. fahr. and showed a large quantity of fog. How much more heat would be needed to obtain a temperature as high as 110 deg. fahr. or more above the vaporization point as was the case of the 440-deg. end-point gasoline to produce a dry mixture? Would it be necessary to go very much higher for kerosene, to get it above the condensation-point?

Mr. Tice:—The vapor temperatures would need to be higher than those given by Professor Wilson or Professor Norman to form a dry charge. Suppose we put liquid fuel in one end of a chamber and draw nothing but vapor from the other end. In its passage through the chamber the fuel is completely vaporized. The vapor outlet is in communication with an air passage. The vapor is saturated; we are vaporizing in a chamber where there is no air. In other words, the vapor-pressure is the whole pressure within it, and the temperature of the liquid must be high enough to support that pressure. When we allow this vapor to pass out and mix with the air, its initial admixture is very incomplete and non-homogeneous. We will get a partial condensation, a liberation of heat, a rise in temperature of the air and a foggy charge; after that we will need to make a further application of heat to dry the charge again because of the imperfections in our means for transferring heat from one substance to the other, or to the liquid particles.

CHAIRMAN HORNING:—Some of the constituents are bound to gather in the form of globules as they go along. As Mr. Tice pointed out, we can burn these globules very efficiently provided they do not become too large. It is difficult to determine just what size is permissible. To obtain a perfectly clear, dry mixture, would require exceedingly high temperatures. Dr. H. C. Dickinson covered that point in his test of the Chalmers engine at the Bureau of Standards, in checking the fuel report that the Society published concerning hot-spots. He observed the temperatures up to 500 deg. fahr. and had liquids running through the manifold even when using a gasoline such as Red Crown. There were some particles that would not stay in the form of vapor at 500 deg. fahr. We would like to have the mixture dry, but there is a point beyond which we do not like to go. I carried on a series of economy tests on one of Mr. Stockfleth's engines for an entire week, to illustrate that. The intake-manifold was entirely surrounded by the exhaust-manifold. The exhaust-manifold was red-hot on the outside; therefore, we had the highest temperature possible.

When Mr. Stockfleth's engine was giving slightly more than 0.570 lb. per b.hp-hr. on gasoline, it would show 0.591 lb. per b.hp-hr. on kerosene. The difference was almost insignificant. During the week's tests the difference in thermal efficiency between gasoline and kerosene was consistently 0.021 lb. per b.hp-hr. The temperatures were high enough to vaporize all the fuel, or at least to get it into such shape that it would do the work. The difference was due merely to some difference in volumetric efficiency, referred to by Mr. Tice. On account of the temperature changes and the vaporization, we got more in with the gasoline than we did with the kerosene; therefore, we obtained a higher thermal efficiency.

MR. TICE:—With the same specific consumption per horsepower-hour, the thermal efficiency is less for kerosene than for gasoline?

CHAIRMAN HORNING:—Kerosene has a higher heat-content than gasoline, and yet we do not get the returns. That is a very important distinction.

MR. TICE:—There is one range of conditions in which we do get an equal or higher thermal efficiency or, in other words, a much less specific fuel-consumption of kerosene than of gasoline; that is, from about one-half load down to the minimum load. It is common to find differences in specific consumption in favor of kerosene of some 6 to 8 per cent.

H. C. GIBSON:—A development in connection with the Knight engine is pertinent to the present discussion. I have contended for many years that we can get more out of fuel by attention to what can be called the alimentary canal of an engine. Engines do not assimilate all the present-day fuel that is fed to them, principally because they do not masticate it for a sufficiently long period.

The inherent rate of vaporization of the fuel is the controlling factor. It has been my belief, and this is now confirmed, that we can evaporate the fuels to a usable condition without introducing liquid fuels into the cylinder. I entirely disagree with any belief in our ability to burn liquid fuel economically, completely or satisfactorily within the cylinder of an automotive engine. I designed and built a car with a 133-in. wheelbase, weighing approximately 5400 lb. loaded, having a rear-axle ratio of 3 10/13 to 1 and a six-cylinder engine of 3¾-in. bore and 6-in. stroke, or 397-cu. in. capacity. It showed remarkable flexibility without changing gears from 2 to 75 m.p.h., without any missing or choking. The fuel consumption on a concrete road at 25 m.p.h. was 1 gal. per 51.4 ton-miles, which is approximately 18.6 miles per gal. According to all the rules, I think an engine of such size in such a car should have shown something less than 12 miles per gal. under ordinary conditions. The crankcase-oil dilution, after 3000 miles of operation without having drained the crankcase, was 8.8 per cent of comparatively heavy distillate, none of which distilled over at less than 410 deg. fahr., the final temperature of the test being

570 deg. fahr. The distillation operation lasted for 90 min., which was time enough to give every opportunity for distilling every drop, including whatever was cracked out of the lubricating oil. It was shown that a very small proportion of true gasoline was in the crankcase oil. Such gasoline as was there comprised only the heaviest ends, and I am rather inclined to doubt that there was any gasoline there at all. Distillation tests run at the standard rate showed no gasoline whatever. The carbon deposit after 3000 miles was really negligible; it could not be measured and could be wiped off. It seemed to have reached a point where it just coated the surfaces, and it did not increase after that first coating had been put on.

The wear of frictional surfaces was not measurable, and there was not a noticeable pound, detonation, or any other disagreeable noise in the engine of any kind under any conditions. In fact, we could not make the engine "ping," although the compression pressure was 88 lb. per sq. in. gage, by three different methods of test, or a total of 103 lb. per sq. in. absolute. It was specifically timed for slow-speed operation, not to exceed in ordinary use 1500 or 1600 r.p.m. All the conditions of that engine are utterly wrong according to general practice, but since they worked out so extraordinarily well they must be worth noting.

This was all attributable to the particular attention paid to and treatment of the alimentary canal of the engine, so to speak. It started at the entrance of the air to the hot oven, which delivered air at a considerable temperature to the carbureter; and such metered mixture of air, gas vapor and liquid as came from the carbureter was inspired into a manifold in which no attempt whatever had been made to maintain the velocity of the gases. In fact, it was just the opposite; the manifold passages were of much greater area than the carbureter-flange passage. It was a plain manifold running straight to each of the cylinders, with a little obstruction to each of the nearer cylinders; that is, more obstruction to cylinders Nos. 3 and 4 than to cylinders Nos. 2 and 5; and a little more obstruction to cylinders Nos. 2 and 5 than to cylinders Nos. 1 and 6. But the main feature of that manifold was very similar to the one mentioned by Mr. Horning, in which the whole of the manifold was jacketed by exhaust gases. The intention was to get as much area as possible available for the evaporation of the liquids, and special attention was paid to and provision made for the catching or trapping of the liquid particles that would be in the manifold under all conditions with the present fuel, and the cooking of those liquid particles off into a gas. That was accomplished by the use of long ribs in the upper and lower halves of the manifold. The manifold was split throughout the length so as to get a perfectly clean job. The ribs, acting like the banks and the bottom of a river and retarding the movement of the gases through the manifold, had the effect of giving a somewhat circulatory motion to the gases traveling along the lower half and, similarly, to those traveling along the upper half. By centrifugal force, that motion threw out some of the liquids that were in suspension in the airstream, so that they were deposited on the rough surfaces of the ribs and were cooked off.

The temperatures of the mixtures going into the cylinders were comparatively low, say 150 to 180 deg. fahr. We could run that engine on any one cylinder after getting it warmed-up, and, so far as we could judge with good test apparatus, we could get practically the same results with any cylinder working with all the others cut out. That showed pretty good distribution.

F. G. SHOEMAKER:—In manifolds in which an attempt is made to supply the heat of vaporization at a very high rate, assuming that heat can be supplied at a very high rate, is there any chance that the fuel will crack if it is in the presence of all the air that is traveling with sufficient velocity to blow it over the liquid fuel?

MR. TICE:—I will not say that it is cracking or what it is, but we do get a deposit in the intake passages. It never amounts to much in thickness; it is rarely found to be more than 0.001 in. thick. One cannot chip enough of it off to measure its thickness accurately, but we do get a deposit which, so far as we can distinguish, is identical with the accumulation at and around the intake-valve.

MR. SHOEMAKER:—In an intake passage that was ribbed in such a manner that an appreciable depth of liquid fuel was caught and held so that air in passing through the intake would not sweep the vapors off and the liquid fuel would boil, would one be more likely to have cracking?

MR. TICE:—Yes, if the intake were arranged to trap liquid, as Mr. Gibson has suggested. Let us consider what happens when we open and close the throttle. We have these extensive accumulations in the intake pipe and any subsequent reduction of pressure in it will cause a higher rate of evolution of vapor from them. The accumulation of liquid will be at its maximum under open-throttle conditions.

When conditions controlling vaporization in such an intake are at their best, we get an excessively rich mixture. Correspondingly, if we provide these pockets, the mixture will be impoverished during the time the pockets are filling. We are taking liquid out of the mixture and, while going from a small throttle-opening to a large one, we will impoverish the mixture in the cylinders. Then, when we close the throttle, we take the liquid out of the pockets.

MR. SHOEMAKER:—Tests were conducted in connection with that point, at Ottawa Beach, in 1920. The conclusion was reached in the report[7] that the rate of acceleration was increased by increasing the intake temperature. It seemed that the obvious point was overlooked, that the improved acceleration was not due to the increased temperature of the intake gases, but to the smaller amount of liquid fuel in the manifold.

MR. TICE:—Upon opening the throttle, the vaporization rate is spoiled as would be the case if much more liquid were in there but, relatively, it is not spoiled so much.

CHAIRMAN HORNING:—Acceleration is the function of the energy content in the mixture at the moment the spark jumps. Therefore, the more fuel in the form of a vapor at the moment the spark jumps, the greater is the energy released. If a large part of the fuel is in liquid form, it will not be released. Dr. Dickinson's demonstration was beautifully illustrative of this point.

MR. TICE:—Fuel in the liquid state does not burn and that is all there is to it.

MR. SHOEMAKER:—In testing out an oil-pressure regulator, I connected a milk bottle as a trap on the vacuum line to the manifold and noticed that the vapor apparently was rushing back-and-forth into the milk bottle from the manifold. I could not believe that it was coming through a length of ¼-in. tubing, and came to the conclusion that the action was due to the pulsation in the manifold, which caused the vapor in the bottle to pass through the dew-point. I believe we see some of the same effect in a manifold when we put in glass windows, and that the momentary waves of high pressure passing back-and-forth through the manifold make us think it is fog, whereas it may be a dry mixture. I have had that occur in manifolds that were bone-dry on the walls, and could not get rid of the apparent fog.

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[7] See TRANSACTIONS, vol. 15, part 2, p. 4.

MR. TICE:—Regardless of what the throttle position is or what the load on the engine is, there is a pulsation pressure in the intake pipe. With the ordinary four-cylinder engine this pressure will run to about 70 mm. (2.756 in.) of mercury. In a four-cylinder engine it varies by about that amount twice per revolution. The change of the total pressure in the intake represents a corresponding change in vapor-pressure. We get exactly what we must expect, in condensation and re-evaporation of the fuel, every time that pressure changes. This is most clearly to be observed when the engine is running at a low speed. If we have a milk bottle or any chamber on the intake, the pulsation in pressure gradually pumps vapor into the bottle and pumps air out until, finally, we have substantially the same mixture in the bottle as in the intake pipe. A small hole merely delays the action. A larger passage allows it to happen earlier. The pressure is changing twice per revolution in that bottle.

W. H. HOLLISTER:—Mr. Tice showed the different openings of the conventional type of throttle and the turbulence in the manifold. Did he ever experiment with a throttle of a different type that would open from the center with a gradually enlarging aperture? If so, what effect would that have on the turbulence and the wet manifold?

MR. TICE:—That has been done. With a conventional butterfly throttle-valve, standing at an angle and being partly open, the greater part of the flow gets past the downstream edge. There is an eddy space behind the throttle where the fluid is revolving. The liquid content of that eddy in an intake will increase to a certain amount but will not become any greater. Liquid is passing into and out of the eddy. With this throttle-valve, except for the content of the eddy, the whole of the passage is being swept by the air.

Suppose we have a throttle that opens up like the iris diaphragm of a camera. The mixture is flowing upward, and we get a symmetrical ring, or doughnut-shaped eddy above the iris. The liquid content of the eddy is relatively greater and the shape does not permit the eddy to elongate; in other words, the eddy remains circular in section, which is the most favorable shape for retaining the maximum amount of liquid. In the butterfly throttle, the eddy includes only a very narrow sectional area and the liquid content of the eddy is less.

In the intake-manifolds I have shown, when the throttle-opening, corresponding to an almost closed iris throttle, was very small and we got a short distance above the throttle plate, the whole cross-sectional area of the pipe was working; but that would not be the case with the iris throttle. We would have then an excessive wall accumulation of liquid and minor eddies clear up the wall until the stream expanded to sweep the walls again.

CHAIRMAN HORNING:—To obtain a distillation diagram, the fuel is put into an Engler flask to which a flame is applied. The temperature of the vapor will assume a definite value which we plot. The curve representing the percentages coming off is known as the distillation curve, which is used generally by the oil industry. If we take Mr. Tice's kerosene and mix it with the very fine unknown substance that he used, some of the heavy particles will come off first; in other words, it does not separate in a well defined way. There is an appreciable quantity of this very heavy end-point in the first distillate that comes off at what is called the initial point. Likewise, when we come to the end-point, we have not yet succeeded in distilling off all this light stuff, although we have been fairly successful. There are some things coming off at the time which we call the end-point, when the material is almost dry. Since temperatures and percentages are involved, a distillation curve is a mathematical expression of a large number of complicated physical factors.

The interesting thing that Professor Wilson has shown is that the temperature of condensation of a 12-to-1 mixture is very important. It proves that the temperature at which Red Crown gasoline will start to condense from a 12-to-1 mixture, which is a rather rich mixture, is far below the point at which the first part of the gasoline comes off in distillation. The standard specification calls for an initial temperature of 140-deg. fahr.; therefore the initial point of condensation is about 20 deg. under the initial boiling-point. This statement holds true for all fuels except those made in Oklahoma from very heavy stuff, and from some of the very light stuff when it is possible to get the condensation point above the initial-point.

If we start a liquid to boiling in an Engler flask, we keep taking off and replacing just the amount that comes off; in other words, what comes off first is mostly the lighter ends, but we are always dragging off some of the heavier ends. Finally, we come to the point where there is no change in temperature; in other words, what is coming off and the liquid in the bowl have the same composition. Consider the drops that rush through a manifold at the rate of 150 per sec., as they often do. The outside layers give off their lighter constituents and finally, as the globules get along far enough, the vapor coming off is the same composition as that of the liquid left on the walls.

To show what distillation means and what the end-point means, suppose we take some of this liquid that has the same composition as the stuff coming off, and distill it. We shall find that it is not this heavy fuel at 30 deg.; that it has an initial-point considerably above this; and that the distillation curve runs parallel and ends 40 deg. higher than the end-point of the fuel. This shows that the end-point of the original distillation was merely a mixture with some lighter fuels. That is very important. When we have talked about end-point, we have thought that we were talking about the heaviest constituents in the fuel. The heaviest fuel is there in large quantities, but not exclusively. The curves of the original fuel and the distillation curves of the equilibrium mixture, all run about parallel.

Professor Wilson has worked out a rule by which, if we take the temperature of the 85-per cent point of the fuel and subtract a constant number of degrees, we get a point at which we have ideal condensation of a 12-to-1 mixture. This is a rough rule and a fairly good one; 243 deg. fahr. from the 85-per cent point would give the distillation point.

That is a very important advance in our information, and it is only because our apparatus is imperfect that we must carry temperatures higher than those of the condensation-points. It, therefore, behooves us to make the very best and most skilful use of surface and of temperature distribution, along the line that Mr. Tice has pointed out, in order to keep these temperatures low; because, it is these temperatures that determine volumetric efficiency, the volumetric efficiency determines the weight of air, the weight of air determines the amount of oxygen, and the amount of oxygen determines the amount of fuel that will burn completely. Therefore, the temperature is a very important thing. The weight of air is one of six or eight factors, but it is the most important factor in determining the horsepower that can be developed in an engine.

MR. TICE:—The weight of air determines the power, other things being equal.

CHAIRMAN HORNING: — Another important thing is that, when the mixture is lean, the amount of energy available depends entirely on the amount of fuel. The moment that one gets the correct proportion, the problem changes. When there is not enough air to neutralize the fuel, we have fuel left over. With the fuel left over, if we go just a reasonable distance beyond, the power will still go up, due to the volume of gas in the cylinder, although the excess liquid does not enter into useful combustion.

MR. TICE:—If we have an engine running on a lean mixture and we progressively enrich the mixture, we progressively increase the power until we arrive at a certain point where any further enrichment of the mixture actually increases the amount of air taken into the engine. The brake mean effective pressure curve plotted against the mixture-ratio starts up; by continually enriching the mixture, the curve becomes flat, then goes up, then becomes flat again. Finally, if enrichment be continued, we reach a point where the temperature is not reduced further and the amount of air taken in is not increased.

The relative importance of the conditions discussed as controlling vaporization, as they exist and can be made to exist in the intake system, is best understood if we think in terms of the rate of vaporization. This latter quantity is solely dependent upon the rate at which vapor is removed from the liquid surface and the rate at which heat is imparted to the liquid. Assuming a given degree of turbulence, or internal motion in the gas and vapor-content of the intake, and a given total pressure, the rate of removal of vapor from the liquid surface is relatively a fixed quantity, except as it varies with the rate of vapor evolution from the liquid. Since this is the case, the general statement can be made that the control of the vaporization rate in the intake is entirely a problem of the ratio of heat input to the quantity of liquid fuel.

Professor Wilson's paper[8] gives a complete and concise statement of the possible conditions of the charge that accompany given intake temperatures. It is based on experimental evidence and is applicable to several typical fuels. But note particularly and do not forget the following comments:

> It must be emphasized at the outset that results secured by such methods represent conditions prevailing at equilibrium. It is possible to obtain "wet" mixtures at temperatures well above the dew-point, because the drops of fuel that are sprayed into the airstream do not have sufficient time to absorb heat and vaporize before the mixture reaches the engine. The results simply indicate what is theoretically obtainable if the time, the degree of atomization, etc., are sufficient to permit equilibrium to be approached fairly closely.
>
> These results indicate clearly that any difficulties in securing complete vaporization of the present commercial gasoline are not due to any inherent limitation in the gasoline itself, or to too low manifold temperatures, once the engine is warmed-up. Indeed, where any serious attempt has been made to heat the incoming gases or the intake-manifold, the temperature attained by the airstream is almost invariably far higher than theoretically necessary. Improvement in vaporization apparently should be secured by better atomization, longer times of contact or by throwing the unvaporized particles out of the insulating airstream onto a hot-spot, rather than by raising the temperature of the mixture as a whole with the attendant disadvantages of this.

The obvious means of accelerating the vaporization rate is an augmented heat-input following the extension of the liquid surface and an increase of the temperature gradient across that surface. Let us first consider how we can extend the liquid surface. Since, ordinarily, the entire wall of the intake passage is well wetted in any case, there is no opportunity to go farther in this direction without building longer passages or putting alcoves on those we have. Then, too, the extension of surface accomplished by this means could be only relatively small and wholly inadequate.

The feasible alternative then in this direction is extension of the liquid surface by finer division or spraying of the fuel. Considering that the sprayed fuel assumes a globular or spherical form, with any reasonable degree of division, it is useful to examine the relations between surface and mass for different possible dimensions of globules, since, other things being equal, this is the relation that determines the rate at which heat can be taken on. The surface of a sphere is equal to πd^2. Its volume, and therefore its relative mass for any single liquid, is equal to $\pi d^3/6$. The surface is as the square and the volume as the cube of the diameter. Setting surface over volume, the relation or ratio $S/V = 6/d$ is found, which shows that the surface-to-volume ratio is inversely as the diameter. To make the case concrete, let us say that one sprayer divides the liquid into globules having a diameter of 0.030 in., just under 1/32 in., and that another sprayer discharges the same amount of liquid but in globules of 0.002-in. diameter. Both of these diameters are entirely reasonable. The smaller globules are 1/15 the diameter of the larger, the surface exposure of the same mass of liquid is 15 times as great in the smaller globules and, if the temperature gradient across the surfaces is the same, the smaller globules will *begin* to vaporize at 15 times the rate of the larger. But as vaporization goes on in these two cases with a fixed temperature-gradient, the difference in rate mounts up enormously and will be thousands of times greater for the smaller globules at the time when their last bits change to the vapor state. But there are very serious difficulties in the way of realizing the foregoing. The smaller the globules diffused through the airstream are, the greater the thermal influence of a globule will be upon its neighbors, the more perfectly they will follow the internal motions in the airstream, and the greater the difficulty is in maintaining a fixed or suitable temperature-gradient from a source of heat.

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[8] See THE JOURNAL, November 1921, p. 313.

If the object sought is a dry or completely vaporized charge, formed in the least possible time, it is necessary largely to prevent diffusion of these small globules into the airstream, and to remove them from the heat-insulating airstream to a place where heat can be taken on nearly equally by each of them, subsequently returning only their vapors to the airstream. The need to separate the globules from the air and to apply heat to them alone follows from the fact that we desire the minimum total heat to be given to the charge. Obviously, if the source of heat supply is accessible to the airstream, the latter will take on heat that will be of little or no use in forwarding vaporization, and will appear as a needlessly high temperature of the charge.

When the fuel is thus separated from the airstream, which it must accompany originally for the sake of metering in the carbureter, the need for a fine division of the liquid increases rather than diminishes, since only by this means can the liquid be distributed with reasonable evenness over the hot surface, and only by this means can the duration of contact be made sufficiently short to prevent any accumulation of the liquid, the undesirable features of which have been discussed earlier.

In an intake system realizing this segregation of very finely divided fuel from the airstream, and its reasonably equal distribution over the surface of the chamber in which it is segregated, heat can be applied almost ad libitum to the walls of the chamber to hasten vaporization, without altering the final charge-temperature. Provided the chamber is neither too small nor insufficiently heated to vaporize the fuel, its size and temperature are without influence upon the temperature of the resulting charge, since the fuel takes on enough heat to vaporize it, and superheating of the vapor is prevented by the presence of and intimate contact with the incoming liquid spray, thus maintaining saturation and equilibrium between the vapor and the liquid.

But even in this case the possible low charge-temperatures discussed by Professor Wilson as supporting the entire fuel in the vapor state are not necessarily realized. When the fuel is separately vaporized and its vapor then mixed with the air, there is a definite temperature-rise in the air following the mixing, depending upon the heat of vaporization of the fuel used, the temperature of the vapor and the mixture-proportion. Therefore, if the entering air has a temperature so low that its value plus this definite temperature-rise upon mixing is less than that required to maintain the fuel as a vapor, some of the fuel will condense. Likewise, if the temperature of the air plus this temperature-rise gives a sum greater than that which maintains all the fuel as vapor, the charge, while entirely gaseous, will unavoidably stand at a temperature higher than that absolutely necessary for dryness.

Average gasoline, when so handled in the intake, causes the charge-temperature to stand at substantially 60 deg. fahr. above that of the entering air, for a 15-to-1 mixture. Thus, to realize Professor Wilson's minimum dry-charge temperature, the entering air will be required to have a temperature of about 33 deg. fahr. Since the temperature of the entering air is almost always higher than this, such an intake system can be taken to give invariably dry charges to the engine.

Professor Wilson's dry-charge temperature for a 15-to-1 kerosene mixture at about atmospheric pressure is given as 230 deg. fahr. In the intake system just discussed, the temperature rise with average kerosene is 125 deg. fahr. The temperature of the entering air would have to be about 105 deg. fahr. to give dryness. Ordinarily, this latter temperature is much less than 105 deg. fahr., with the result that some of the kerosene vapor is condensed upon mixing with the air. Professor Wilson's charts show to what extent this condensation will occur with different temperatures of the entering air.

MOTOR-TRANSPORT PERFORMANCE WITH VARIED FUEL-VOLATILITY[1]

By C T Coleman[2]

The paper is a presentation of data obtained regarding the economic operation of motor vehicles with fuels of varied volatility, under service conditions. The entire fleets of motor-transportation units used by the Post Office Department in Philadelphia and in Pittsburgh were made available for these tests, the resulting data being summarized in the tables and charts and commented upon briefly.

The test procedure is described and the method of examining the used-oil samples for the percentage of dilution, changes in viscosity, flash-point and fire-test is explained, inclusive of the tabulated results.

While the results may not hold true for cold-weather operation, they show that the number of car-miles per barrel of crude oil increases as the volatility of the fuel decreases, over the range of fuels used. This is true because the percentage of production per barrel of crude oil is increased, while the number of car-miles per gallon of fuel is not affected materially by the variation in the volatility as represented by the four fuels used.

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[1] Semi-Annual Meeting paper.

[2] M.S.A.E.—Associate mechanical engineer, Bureau of Standards, City of Washington.

In order that natural resources may be conserved and the ever-increasing number of motor vehicles supplied with fuel, every effort has been made in the last few years to refine all the gasoline possible from each barrel of crude oil and still furnish a satisfactory fuel. If a fuel of higher end-point, or lower volatility, could be used economically, gasoline production could be increased, thus postponing possible future shortage.

TABLE 1—CLASSIFICATION OF THE FLEET UNITS

Philadelphia			Pittsburgh		
Number of Vehicles	Size, Tons	Make	Number of Vehicles	Size, Tons	Make
37	3/8	Ford	42	3/4	White
82	3/4	G.M.C.	16	2	White
31	1 1/2	G.M.C.	16	3	Riker
13	3	Packard	....	....	.....

To the end that some information might be available regarding the economic operation of motor vehicles with fuels of various volatilities it was believed that a series of tests made under service conditions would furnish valuable data.

A request was therefore made of the Post Office Department for cooperation in a test of this kind. That Department readily agreed and placed at the disposal of the Bureau of Standards, for the purpose of test, its entire fleets of motor-transportation units in Pittsburgh and in Philadelphia. Much credit is due the Post Office Department and the personnel at Pittsburgh and at Philadelphia for the manner in which these tests were conducted. The personnel of the fleets in both these cities made every effort to cooperate with the Bureau of Standards' representative, and the results obtained are a tribute to the efficiency of the personnel conducting it. The combined fleets represented some 237 units of transportation, classified as shown in Table 1. The Philadelphia fleet consumes approximately 1000 gal. of fuel per day, while the fleet in Pittsburgh requires approximately 700 gal. of fuel per day. This combination presented two excellent features; first, a rather large number of trucks well divided in respect to tonnage and make; second, two distinct types of operating conditions, inasmuch as Philadelphia has comparatively smooth streets with few hills, while in Pittsburgh the majority of the streets have a very rough surface and there are many hills.

After a conference with representatives of the American Petroleum Institute, the Society of Automotive Engineers, the Bureau of Mines and the refiners, it was decided that, for comparative purposes, the same four fuels should be used in this test as were to be used in the Society of Automotive Engineers' Cooperative Fuel Tests. These fuels are styled Grades *A, B, C* and *D*. Grade-*B* fuel represents the average commercial gasoline on the market at present and is used as a comparative or reference fuel. Grade-*A* fuel is of higher volatility, or lower end-point. Grades-*C* and *D* fuels represent lower volatility and higher end-point in the order of their alphabetical designation.

The average distillation-curves for the fuels used are shown in Fig. 1. Compared to Grade-*B* fuel, these fuels

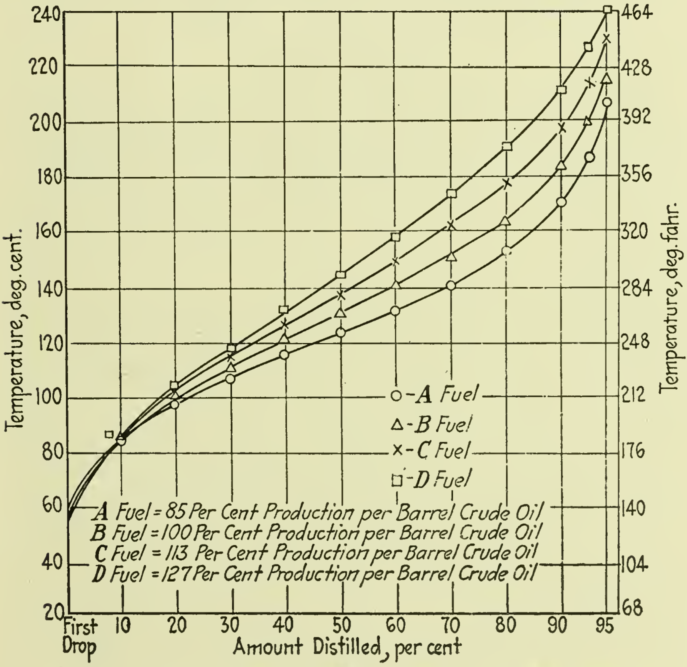

FIG. 1—DISTILLATION CURVES OF THE FOUR TEST-FUELS

represent the average production per barrel of crude oil shown in Table 2.

TABLE 2—ESTIMATED PRODUCTION OF THE FOUR TEST-FUELS

Refiners	Grade *A*, Per Cent	Grade *B*, Per Cent	Grade *C*, Per Cent	Grade *D*, Per Cent
1	81	100	114	128
2	91	100	111	124
3	83	100	113	122
4	84	100	115	133
Average.....	85	100	113	127

In testing these fuels the following relative information was sought:

(1) Miles per gallon compared to Grade-*B* fuel
(2) Crankcase dilution in the engines run on these fuels as compared to dilution using Grade-*B* fuel
(3) Performance in service, particularly as to starting of engines, idling and general operation

TEST PROCEDURE

The following methods of test were used: The transportation units in Pittsburgh and in Philadelphia were each divided into two equal parts or sections designated as Section No. 1 and Section No. 2. In dividing the units, one-half of the total number of each make of truck was placed in each section. This method gave two sections equal in respect to the number of vehicles as well as to the capacities and makes.

One section of the fleet was at all times operated on Grade-*B* fuel in order that differences due to changes in weather conditions might be noted and allowed for in the final comparisons. Each truck in Section No. 1 was designated by the numeral 1, placed in some convenient place to be recognized by the truck dispatcher and the men putting fuel into the tanks. The same method was applied to Section No. 2, where the numeral 2 was placed.

From each section, three trucks of each make were selected; the crankcases were drained of used oil and new oil was put in each week or period of the test. At the end of each run or each week a 1-qt. sample was drained from each engine thus selected and labeled as to the truck number, the make, the kind of fuel and the

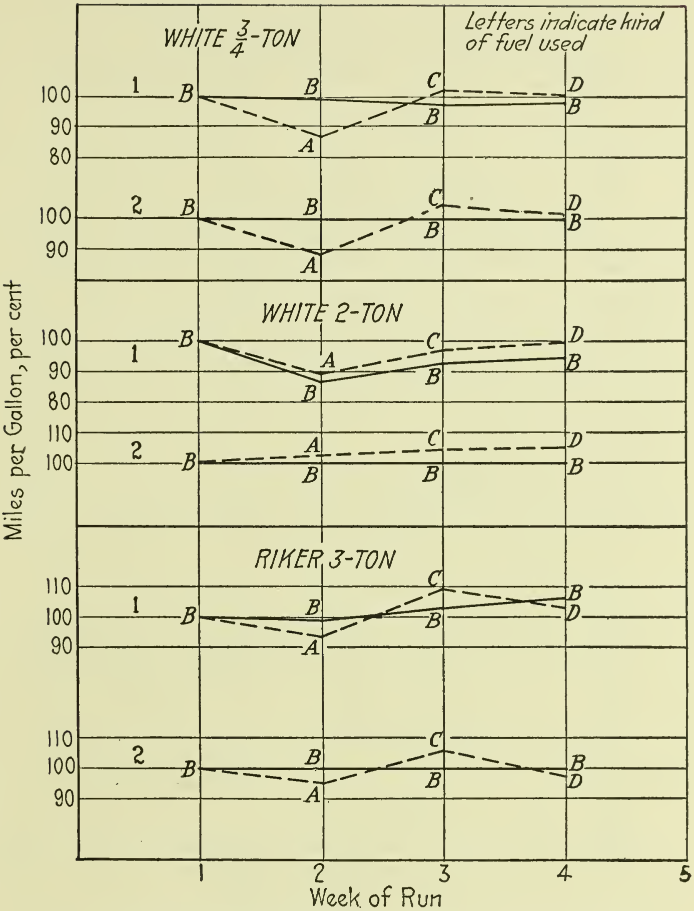

FIG. 2—CHART SHOWING THE MILEAGE PER GALLON OBTAINED AT PITTSBURGH WITH DIFFERENT MAKES OF VEHICLE

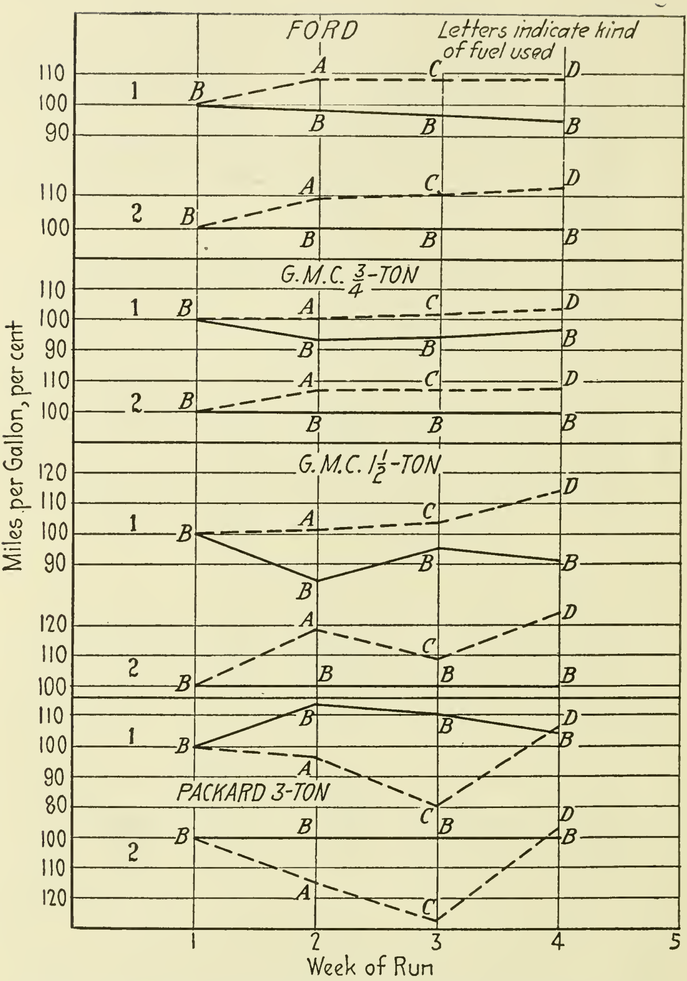

FIG. 3—CHART SHOWING THE MILEAGE PER GALLON OBTAINED AT PHILADELPHIA WITH DIFFERENT MAKES OF VEHICLE

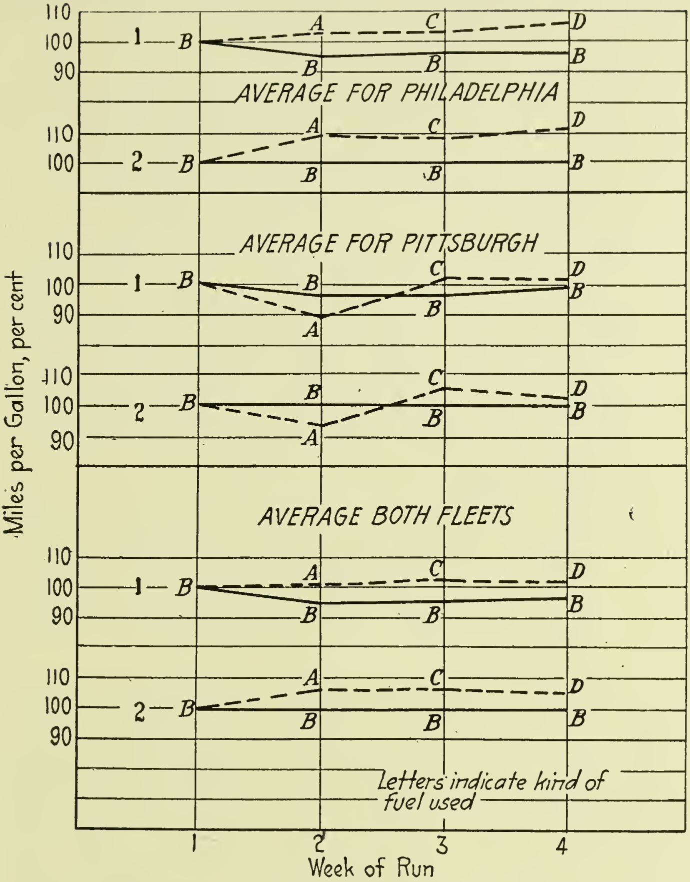

Fig. 4—Average Mileage per Gallon of Fuel for Each Week of the Test at Philadelphia and Pittsburgh and the General Average for Both Fleets

mileage. Each truck was labeled with the letter corresponding to the kind of fuel on which it was operating. This letter was placed in such a manner that mistakes were unlikely in placing the proper fuel in the tank.

The schedule of test was as follows:

First Week—Both sections were run on Grade-*B* fuel. All trucks were drained, the fuel was placed in the tank and the mileage was noted. At the end of the week the fuel remaining in the tanks was drained and measured, and this amount was deducted from the total fuel put into the tanks during the run. The mileage was noted at the end of a run

Second Week—Section No. 1 continued on Grade-*B* fuel and Section No. 2 was run on Grade-*A* fuel; the same procedure was followed as for the first week.

Third Week—Section No. 1 continued on Grade-*B* fuel and Section No. 2 was run on Grade-*C* fuel, following the same procedure as in the two previous runs

Fourth Week—Section No. 1 was again continued on Grade-*B* fuel, and Section No. 2 was run on Grade-*D* fuel, following the same procedure as in the preceding weeks

As each shipment of fuel was placed in the supply tanks, a 1-gal. sample was taken and labeled as to the date and the kind of fuel. Distillations were run on these samples to check the fuel delivered by the refinery. All samples were shipped to the Bureau of Standards at the City of Washington, where the laboratory tests were made

Table 3 gives the individual truck averages for each week of the 4-week runs. A wide variation will be noted in miles per gallon for trucks of the same make, which may be explained by differences in the engine condition and adjustment, the type of service and the operators. Table 3 also gives the duties of the trucks in Philadelphia. The low mileage on some of the trucks used in letter-box collection-service can be understood better when it is realized that these trucks make an average of some 300 stops in the course of an 8-hr. tour of duty. The trucks in Pittsburgh are required to perform service similar to that of those in Philadelphia, except that the ¾-ton White trucks used in Pittsburgh for letter-box collection-service do not make so many stops in a tour of duty as do the Fords in Philadelphia. Table 4 gives the average consumption of other fuels by individual trucks in both fleets compared with that of Grade-*B* fuel.

TABLE 3—TEST RUNS IN PHILADELPHIA FOR 4 WEEKS ON THE ECONOMIC VOLATILITY OF FUELS

Section No. 1						Section No. 2					
Car No.	Service Rendered	Fuel Consumption, Miles per Gal.				Car No.	Service Rendered	Fuel Consumption, Miles per Gal.			
		First Week Grade *B*	Second Week Grade *B*	Third Week Grade *B*	Fourth Week Grade *B*			First Week Grade *B*	Second Week Grade *A*	Third Week Grade *C*	Fourth Week Grade *D*
Ford Cars						Ford Cars					
500	(1)	N.S.	10.19	11.13	13.35	1,708	(5)	13.84	13.54	13.31	7.50
501	(2)	9.73	14.95	7.42	7.12	1,710	(5)	9.06	7.51	7.80	9.49
502	(3)	7.16	9.38	7.20	N.S.	1,711	(5)	12.31	12.04	11.26	10.83
503	(1)	N.S.	12.38	11.43	11.23	1,712	(5)	11.97	17.24	17.18	14.18
504	(1)	8.43	8.73	7.62	7.66	1,714	(5)	13.80	12.14	13.40	14.44
505	(2)	9.58	8.44	8.70	7.22	1,715	(5)	8.00	8.20	10.34	10.37
506	(3)	8.88	12.37	9.88	4.60	1,716	(5)	7.71	7.89	N.S.	N.S.
509	(1)	7.83	8.26	9.02	8.47	1,717	(5)	10.56	11.90	11.84	12.54
510	(3)	11.36	9.76	8.37	8.32	1,718	(5)	8.41	9.08	9.67	9.90
511	(3)	7.30	6.14	8.00	11.69	1,719	(5)	9.96	9.72	10.15	9.26
512	(2)	8.71	7.42	9.61	8.93	1,720	(5)	5.40	11.58	14.08	12.84
513	(1)	14.54	11.27	11.30	11.09	1,721	(5)	8.95	9.31	8.72	8.96
514	(3)	8.84	8.00	N.S.	6.59	1,722	(5)	10.87	12.81	9.88	11.58
515	(3)	9.82	N.S.	N.S.	N.S.	1,723	(5)	8.53	7.44	8.77	9.40
846	(2)	7.25	6.71	6.48	6.74	1,724	(5)	12.32	12.91	9.06	10.79
847	(3)	8.31	8.62	8.23	9.16	1,725	(5)	7.74	9.76	11.26	11.30
850	(2)	6.22	4.83	6.10	9.03	1,726	(5)	11.77	13.16	11.63	11.11
1,729	(5)	12.48	11.42	11.93	10.51	1,727	(5)	9.98	9.52	8.37	9.46
S-100	(4)	11.83	6.25	10.29	8.67	1,728	(12)	8.32	9.11	7.09	10.62
¾-Ton G. M. C. Trucks						¾-Ton G. M. C. Trucks					
2,663	(6)	7.82	10.41	7.92	7.00	2,332	(7)	10.69	10.82	13.51	10.94
2,664	(7)	7.62	8.54	9.43	9.17	2,333	(7)	8.91	10.34	10.51	10.00
2,665	(7)	9.20	8.97	8.06	11.60	2,334	(7)	5.77	6.98	8.22	11.11
2,666	(6)	8.99	9.72	9.80	9.44	2,335	(7)	10.86	8.08	7.45	6.58
2,667	(6)	11.10	8.99	8.77	8.68	2,336	(3)	9.92	6.97	6.14	5.45
2,668	(6)	9.98	11.79	12.53	13.59	2,419	(7)	6.29	6.36	9.54	7.50
2,669	(6) and (8)	8.07	8.29	8.44	9.63	2,422	(7)	5.93	5.79	6.18	5.57
2,670	(8)	5.74	5.81	6.89	6.50	2,423	(7)	7.36	6.32	4.40	7.63
2.671	(8)	8.76	5.50	6.40	5.33	2,424	(7)	7.17	7.02	7.39	7.28
2,672	(6)	7.10	6.35	6.06	6.00	2,425	(3)	N.S.	6.70	3.41	8.60
2,673	(8)	11.44	5.42	6.02	5.73	2,426	(3)	7.35	N.S.	6.67	5.85
2,674	(6)	9.91	7.30	6.65	7.06	2,427	(3)	9.46	8.52	8.48	9.00
2.675	(6)	9.75	6.94	7.93	9.19	2,428	(3)	4.67	4.56	10.47	5.63
2,676	(1)	5.80	4.28	5.64	4.96	2,429	(7)	9.04	8.62	8.94	9.84
2,677	(2)	8.10	9.27	8.91	8.14	2,430	(7)	6.94	6.70	7.50	7.46
2,678	(8)	3.08	8.51	7.95	5.70	2,431	(7)	7.02	8.82	6.94	9.02
2,679	(2)	5.48	7.74	7.72	8.31	2,432	(7)	4.71	4.15	5.41	5.14
2,680	(7)	8.50	8.61	10.23	8.68	2,433	(3)	8.99	8.09	2.73	5.50
2,681	(7)	5.73	5.19	5.02	6.05	2,434	(7)	N.S.	11.09	10.85	12.83
2,682	(7)	5.99	5.92	5.73	5.40	2,435	(7)	7.71	5.19	5.60	4.51
2,683	(6)	6.72	6.69	6.98	6.64	2,436	(7)	5.79	6.54	7.27	8.80
2,684	(7)	5.67	5.00	4.84	5.12	2,446	(8)	7.11	9.72	9.46	8.41
2,685	(7)	4.42	5.73	6.34	5.97	2,450	(8)	10.29	10.27	10.71	12.29
2,686	(7)	6.76	5.21	6.01	6.31	2,451	(8)	10.17	10.90	11.21	10.91
2,687	(6)	7.87	7.07	7.22	7.13	2,452	(8)	4.07	4.79	5.22	3.69
2,688	(6)	7.64	9.01	4.87	6.55	2,453	(8)	5.59	5.22	5.66	5.07
2,689	(6)	7.64	5.55	4.67	6.46	2,454	(10)	10.89	8.68	9.75	8.91
2,690	(7)	6.47	4.89	5.67	7.39	2,455	(8)	9.95	8.20	9.32	11.00
2,691	(1)	4.10	6.62	6.23	7.41	2,456	(8)	5.88	6.00	6.48	6.61
2,692	(6)	8.48	7.50	5.46	8.28	2,457	(8)	7.17	7.36	6.80	6.52
2,693	(7)	6.20	6.11	6.02	6.06	2,458	(8)	11.56	12.32	10.27	9.26
2,694	(7)	8.04	6.55	6.35	5.43	2,459	(3)	5.19	5.60	5.88	6.76
2,696	(7)	9.59	8.06	8.83	8.00	2,460	(7)	6.48	6.32	9.12	8.93
2,697	(7)	10.51	7.48	9.84	8.77	2,461	(1)	5.48	5.90	5.14	4.04
2,698	(7)	8.42	7.48	7.19	7.40	2,462	(1)	7.69	8.56	9.17	9.86
2,699	(7)	9.42	10.17	9.00	8.59	2,463	(2)	8.85	9.48	7.18	6.94
2,700	(7)	6.95	6.52	6.12	5.65	2,464	(2)	6.21	6.78	10.13	8.11
2,717	(7)	8.37	6.83	9.00	7.32	2,465	(6)	10.93	14.33	10.74	10.51
2,718	(7)	7.56	5.28	5.63	6.96	2,466	(7)	7.96	11.42	9.51	10.71
2,719	(3)	8.18	7.40	7.97	8.03	2,660	(7)	9.21	7.88	8.15	7.83
2,720	(3)	9.17	7.02	8.94	9.66	2,661	(3)	8.02	6.00	7.26	7.47
.....	....	......	......	......	......	2,662	(6) and (8)	6.42	5.66	6.95	9.34
1½-Ton G. M. C. Trucks						1½-Ton G. M. C. Trucks					
3,033	(7)	8.55	5.81	5.97	6.13	4,304	(6)	6.56	4.22	5.85	7.21
4,303	(9)	9.01	5.71	5.29	8.70	4,309	(8)	5.86	5.44	4.21	4.84
4,311	(3)	6.73	7.30	5.84	5.38	4,314	(8)	7.77	5.31	5.85	8.13
4,315	(8)	6.21	4.11	8.11	6.09	4,317	(3)	4.49	4.77	6.16	5.79
4,318	(8)	3.41	3.14	3.12	3.16	4,319	(6)	6.33	5.74	6.93	8.38
4,321	(8)	3.62	4.12	3.60	4.53	4,330	(8)	4.72	5.40	5.84	6.32
4,437	(3)	6.33	N.S.	N.S.	N.S.	4,851	(8)	5.21	5.53	6.61	4.32
4,440	(3)	6.00	4.00	N.S.	N.S.	4,853	(8)	3.74	4.01	3.96	4.67
4,442	(8)	6.03	4.03	4.46	4.57	4,855	(8)	5.05	6.55	4.63	5.14
4,852	(8)	4.00	3.92	4.84	4.67	4,857	(8)	5.24	5.04	4.58	4.18
4,854	(8)	3.88	5.29	5.46	5.25	5,011	(8)	4.43	6.26	6.12	4.94
4,856	(8)	7.15	6.32	6.16	6.93	5,013	(8)	4.79	5.22	5.80	5.08
4,858	(8)	2.66	4.33	4.32	3.55	5,015	(8)	4.03	3.62	3.23	3.96
5,012	(10)	3.95	3.11	4.00	4.68	5,017	(8)	6.74	5.27	5.99	8.44
5,014	(8)	6.60	4.89	4.19	4.63	S-402	(4)	3.92	6.99	6.34	8.62
5,016	(3)	6.10	5.57	9.67	4.29	......	....	......	......	......	......
3-Ton Packard Trucks						3-Ton Packard Trucks					
6,168	(8)	2.34	2.53	4.83	2.67	6,169	(8)	3.19	2.84	3.16	4.70
6,301	(8)	2.77	4.75	2.90	3.59	6,302	(8)	3.04	5.18	3.39	5.06
6,303	(8)	3.77	3.97	3.32	3.11	6,420	(8)	3.47	3.46	3.09	3.11
6,421	(8)	3.13	3.06	3.88	3.80	6,422	(8)	3.29	3.02	3.16	3.47
6,423	(8)	2.86	2.86	2.60	2.70	6.424	(8)	5.02	3.35	1.94	2.75
6,425	(8)	2.61	2.64	2.61	2.31	6,493	(3)	4.00	N.S.	......	4.35
6,495	(8)	3.03	3.54	2.52	3.03	......	....	......	......	......	......

(1) *Pouch Service*—Carrying pouches of first-class, special and registered mail, from the Central Post Office to and from the various sub-stations. The trucks are handicapped by the congested conditions in the alleyway at the Central Post Office.
(2) *Package-Box Collections*—Collection of mail from package boxes.
(3) *Reserve*—Extra cars placed in service when regular trucks are disabled or when the volume of mail to be transported is excessive.
(4) *Service Car*—Stationed at the Central Post Office for truck repair there; responds to service calls also.
(5) *Letter-Box Collection*—Collecting mail from letter-boxes.
(6) *Interstation*—Carrying mail between the Central Post Office and sub-stations; same handicap as for (1).
(7) *Parcel Post*—Delivery of parcel-post mail.
(8) *Screen-Wagon*—Hauling mail between the Central Post Office and the railroad depot; same handicap as for (1).
(9) *Equipment and Repair Truck*—Used for repairing letter-boxes, box-posts and the like.
(10) *Registry Truck*—Hauls registered mail between the Central Post Office and the Pennsylvania Railroad depot.
(11) *Registry Car*—Hauls registered mail to the banks.
(12) *Truck Supervision.*

TABLE 4—INDIVIDUAL-VEHICLE AND FLEET AVERAGES FOR FUEL CONSUMPTION

		Philadelphia Fleet										Pittsburgh Fleet								Average for Both Fleets	
		Ford		¾-Ton G.M.C. Truck		1½-Ton G.M.C. Truck		3-Ton Packard Truck		Fleet Average		¾-Ton White Truck		2-Ton White Truck		2-Ton Riker Truck		Fleet Average			
		Section No. 1	Section No. 2	Section No. 1	Section No. 2	Section No. 1	Section No. 2	Section No. 1	Section No. 2	Section No. 1	Section No. 2	Section No. 1	Section No. 2	Section No. 1	Section No. 2	Section No. 1	Section No. 2	Section No. 1	Section No. 2	Section No. 1	Section No. 2
First Week	Grade of Fuel	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*	*B*
	Fuel Consumption, miles per gal.	9.31	9.97	7.71	7.74	5.64	5.26	2.92	3.66	7.22	7.50	5.61	5.81	5.34	4.77	2.37	2.46	4.89	4.78	6.53	6.65
	Correction for Condition of Fleet, per cent	....	-6.7	....	-0.3	....	+7.2	....	-20.2	....	-3.8	....	-3.4	....	+12.0	....	-3.7	....	+3.3	....	-1.8
Second Week	Grade of Fuel	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*A*	*B*	*A*
	Fuel Consumption, miles per gal.	9.17	10.78	7.21	7.78	4.77	5.29	3.34	3.57	6.86	7.76	5.57	5.09	4.68	4.27	2.33	2.31	4.72	4.29	6.21	6.71
	Correction for Condition of Fleet, per cent	....	10.06	....	7.75	....	5.67	....	2.85	....	7.48	....	4.91	....	4.78	....	2.23	....	4.39	....	6.59
	Adjusted Values to First-Week Basis	9.31	10.23	7.71	8.29	5.64	6.70	2.92	2.49	7.22	7.88	5.61	4.94	5.34	5.46	2.37	2.27	4.89	4.55	6.53	6.93
	Change in Mileage, Grade *B* to Grade-*A* Fuel, per cent	....	+9.8	....	+7.5	....	+18.8	....	-14.7	....	+9.0	....	-12.0	....	+2.2	....	-4.2	....	-7.0	....	+6.1
Third Week	Grade of Fuel	*B*	*C*	*B*	*C*	*B*	*C*	*B*	*C*	*B*	*C*	*B*	*C*	*B*	*C*	*B*	*C*	*B*	*C*	*B*	*C*
	Fuel Consumption, miles per gal.	8.98	10.77	7.29	7.89	5.36	5.47	3.24	2.95	6.95	7.78	5.43	5.88	4.92	4.61	2.44	2.69	4.72	4.86	6.27	6.84
	Correction for Condition of Fleet, per cent	....	10.05	....	7.86	....	5.86	....	2.36	....	7.49	....	5.67	....	5.16	....	2.59	....	4.97	....	6.72
	Adjusted Values to First-Week Basis	9.31	10.41	7.71	8.31	5.64	6.16	2.92	2.13	7.22	7.78	5.61	5.86	5.34	5.60	2.37	2.52	4.89	5.15	6.53	7.00
	Change in Mileage, Grade *B* to Grade-*C* Fuel, per cent	....	+11.8	....	+7.7	....	+9.3	....	-27.1	....	+7.7	....	+4.3	....	+4.8	....	+6.1	....	+5.2	....	+7.1
Fourth Week	Grade of Fuel	*B*	*D*	*B*	*D*	*B*	*D*	*B*	*D*	*B*	*D*	*B*	*D*	*B*	*D*	*B*	*D*	*B*	*D*	*B*	*D*
	Fuel Consumption, miles per gal.	8.85	10.81	7.44	8.03	5.18	6.00	3.07	3.90	6.95	7.97	5.51	5.81	5.04	4.75	2.51	2.55	4.83	4.82	6.29	6.79
	Correction for Condition of Fleet, per cent	....	10.09	....	8.00	....	6.43	....	3.11	....	7.68	....	5.61	....	5.32	....	2.46	....	4.93	....	6.67
	Adjusted Values to First-Week Basis	9.31	10.61	7.71	8.29	5.64	7.00	2.92	2.96	7.22	7.98	5.61	5.71	5.34	5.64	2.37	2.32	4.89	4.99	6.53	6.92
	Change in Mileage, Grade *B* to Grade-*D* Fuel, per cent	....	+13.9	....	+7.5	....	+23.1	....	+1.3	....	+10.5	....	+1.8	....	+5.6	....	-1.9	....	+2.1	....	+6.0

TABLE 5—MISCELLANEOUS DATA ON OIL USED

	Philadelphia Samples													
Period	Kind of Oil	Make of Truck	Truck No.	Number of Miles	Grade of Fuel	Section No.	Initial Point		Percentage Distilled at 300 Deg. Cent. (572 Deg. Fahr.)	Gravity		Viscosity at 100 Deg. Fahr., Saybolt Sec.	Flash-Point, Deg. Fahr.	Fire-Test, Deg. Fahr.
							Deg. Cent.	Deg. Fahr.		Specific	Baumé			
First Week	New Medium	....	....	....	....	....	(*a*)300	(*a*)572.0	......	0.906	24.5	316	415	475
	Used Medium	Ford	850	143	*B*	1	163	325.4	5.0	0.905	24.7	293	285	440
			512	244	*B*	1	165	329.0	6.0	0.900	25.6	280	240	390
			513	349	*B*	1	182	359.6	4.0	0.902	25.2	298	315	435
			1,711	308	*B*	2	185	365.0	3.5	0.904	24.9	320	315	450
			1,708	350	*B*	2	168	334.4	4.0	0.902	25.2	294	325	440
			1,710	253	*B*	2	180	356.0	4.5	0.902	25.2	299	305	410
		¾-Ton G.M.C.	2,691	127	*B*	1	171	339.8	6.5	0.900	25.6	242	255	245
			2,679	284	*B*	1	178	352.4	7.0	0.899	25.7	247	235	345
			2,692	352	*B*	1	188	370.4	11.0	0.895	26.4	170	(*b*)	(*b*)
			2,463	433	*B*	2	180	356.0	9.5	0.898	25.9	208	215	360
			2,466	225	*B*	2	179	354.2	7.5	0.902	25.2	258	225	445
			2,458	352	*B*	2	148	298.4	9.0	0.899	25.7	203	210	415
	New Heavy	....	....	....	....	....	300	572.0	......	0.931	20.4	486	430	500
	Used Heavy	1½-Ton G.M.C.	4,318	109	*B*	1	165	329.0	8.0	0.898	25.9	249	215	345
			3,033	201	*B*	1	175	347.0	8.0	0.899	25.7	234	210	320
			4,858	145	*B*	1	168	334.4	4.0	0.901	25.4	315	275	435
			4,304	240	*B*	2	148	298.4	11.0	0.893	26.8	167	175	255
			4,851	213	*B*	2	160	320.0	7.0	0.898	25.9	230	230	370
			4,319	242	*B*	2	135	275.0	5.5	0.900	25.6	247	210	410
		3-Ton Packard	6,425	141	*B*	1	170	338.0	6.5	0.898	25.9	200	215	360
			6,303	130	*B*	1	173	343.4	8.5	0.898	25.9	260	210	320
			6,168	114	*B*	1	176	348.8	3.5	0.901	25.4	265	235	415
			6,302	129	*B*	2	155	311.0	3.5	0.902	25.2	326	240	455
			6,169	136	*B*	2	150	302.0	14.5	0.890	27.3	153	130	185
			6,434	197	*B*	2	158	316.4	8.0	0.900	25.6	284	195	330
Second Week	New Medium	....	....	....	....	....	(*a*)300	(*a*)572.0	......	0.906	24.5	316	415	475
	Used Medium	Ford	513	355	*B*	1	175	347.0	3.0	0.900	25.5	266	320	455
			512	245	*B*	1	150	302.0	7.5	0.898	25.9	212	220	375
			850	217	*B*	1	170	338.0	8.0	0.897	26.0	211	225	405
			1,710	311	*A*	2	152	305.6	4.5	0.903	25.0	300	270	450
			1,711	287	*A*	2	156	312.8	3.0	0.903	25.0	303	335	475
			1,708	357	*A*	2	167	332.6	2.5	0.902	25.2	294	255	450
		¾-Ton G.M.C.	2,679	282	*B*	1	162	323.6	8.5	0.897	26.1	206	205	350
			2,691	139	*B*	1	175	347.0	5.5	0.900	25.5	240	255	400
			2,692	352	*B*	1	160	320.0	11.0	0.893	26.7	181	200	285
			2,463	417	*A*	2	156	312.8	7.5	0.897	26.1	210	205	395
			2,466	191	*A*	2	130	266.0	6.0	0.899	25.7	238	225	410
			2,458	355	*A*	2	150	302.0	7.0	0.897	26.1	221	205	430
	New Heavy	....	....	....	....	....	....	....	......	0.931	20.4	486	430	500
	Used Heavy	1½-Ton G.M.C.	3,033	186	*B*	1	....	....	8.5	0.897	26.1	231	185	350
			4,858	145	*B*	1	168	334.4	4.0	0.900	25.6	279	230	450
			4,318	113	*B*	1	163	325.4	7.0	0.898	25.9	233	210	350
			4,304	227	*A*	2	155	311.0	8.0	0.898	25.9	194	185	335
			4,851	36	*A*	2	160	320.0	2.5	0.892	26.9	268	265	460
			4,319	241	*A*	2	135	275.0	4.5	0.899	25.7	243	210	455
		3-Ton Packard	6,303	119	*B*	1	157	314.6	4.5	0.901	25.4	253	230	455
			6,425	116	*B*	1	150	302.0	5.0	0.898	25.9	207	210	415
			6.168	96	*B*	1	157	314.6	4.0	0.901	25.4	254	240	455
			6,424	137	*A*	2	148	298.4	5.5	0.898	25.9	225	200	405
			6,169	129	*A*	2	145	293.0	13.0	0.888	27.7	123	145	190
			6,302	156	*A*	2	152	305.6	3.0	0.901	25.4	255	230	460
Third Week	Used Medium	Ford	850	177	*B*	1	(*c*)	.....	4.0	0.903	25.0	337	350	460
			512	245	*B*	1	168	334.4	6.5	0.899	25.7	171	230	400
			513	356	*B*	1	163	325.4	5.0	0.900	25.4	260	285	455
			1,711	287	*C*	2	186	366.8	6.0	0.901	25.3	248	280	410
			1,710	238	*C*	2	175	347.0	6.5	0.901	25.3	267	255	385
			1,708	407	*C*	2	190	374.0	6.0	0.901	25.3	241	270	400
		¾-Ton G.M.C.	2,692	362	*B*	1	152	305.6	8.5	0.898	25.9	210	200	390
			2,691	134	*B*	1	176	348.8	5.0	0.899	25.7	241	255	440
			2,679	278	*B*	1	167	332.6	5.0	0.899	25.7	267	255	435
			2,458	368	*C*	2	168	334.4	18.5	0.885	28.2	115	170	225
			2,463	412	*C*	2	170	338.0	17.0	0.890	27.2	134	190	235
			2,466	270	*C*	2	183	361.4	13.0	0.894	26.6	165	210	285
	Used Heavy	1½-Ton G.M.C.	3,033	200	*B*	1	....	.....	8.0	0.898	25.9	235	205	385
			4,318	103	*B*	1	150	302.0	7.5	0.897	26.0	259	185	300
			4,858	136	*B*	1	....	.....	3.5	0.901	25.4	293	270	445
			4,304	240	*C*	2	160	320.0	13.5	0.891	27.1	175	180	245
			4,319	245	*C*	2	160	320.0	9.0	0.898	25.9	209	210	340
			4,851	108	*C*	2	173	343.4	6.5	0.898	26.2	230	230	400
		3-Ton Packard	6,303	123	*B*	1	151	303.8	4.0	0.901	25.4	314	230	475
			6,425	115	*B*	1	147	296.6	4.5	0.899	25.7	255	215	235
			6,168	140	*B*	1	164	327.2	4.0	0.901	25.4	315	225	455
			6,424	69	*C*	2	170	338.0	9.0	0.894	26.6	178	195	285
			6,169	143	*C*	2	155	311.0	30.0	0.871	30.7	77	140	175
			6,302	145	*C*	2	172	341.6	6.5	0.899	25.7	248	220	445
Fourth Week	Used Medium	Ford	850	158	*B*	1	....	.....	3.0	0.907	24.3	339	345	465
			512	259	*B*	1	107	224.6	4.0	0.904	24.9	325	290	445
			513	294	*B*	1	187	368.6	2.5	0.903	25.0	306	375	455
			1,710	309	*D*	2	205	401.0	8.0	0.898	25.9	252	260	400
			1,711	287	*D*	2	205	401.0	6.5	0.900	25.5	261	280	395
			1,708	....	....	....	....	.....	......	.....	....	....	....	....
		¾-Ton G.M.C.	2,679	249	*B*	1	175	347.0	4.0	0.902	25.2	306	260	435
			2,692	298	*B*	1	172	341.6	7.5	0.899	25.7	229	210	375
			2,691	122	*B*	1	....	.....	5.5	0.899	25.7	267	255	440
			2,463	378	*D*	2	185	365.0	30.5	0.876	29.8	84	180	205
			2,458	287	*D*	2	170	338.0	36.0	0.874	30.2	71	255	440
			2,466	182	*D*	2	....	.....	19.0	0.891	27.1	149	215	250
	Used Heavy	3-Ton Packard	6,425	83	*B*	1	....	.....	5.0	0.899	25.7	287	215	445
			6,168	131	*B*	1	174	345.2	4.0	0.901	25.4	333	245	480
			6,303	101	*B*	1	162	323.6	3.5	0.903	25.0	350	255	475
			6,302	119	*D*	2	178	352.4	6.0	0.898	25.9	272	220	385
			6,424	114	*D*	2	175	347.0	18.5	0.886	28.0	138	190	215
			6,169	127	*D*	2	160	320.0	31.0	0.872	30.5	82	145	190
		1½-Ton G.M.C.	4,858	117	*B*	1	174	345.2	3.0	0.902	25.2	330	275	460
			4,318	109	*B*	1	....	.....	6.5	0.900	25.5	285	220	450
			3,033	98	*B*	1	168	334.4	7.0	0.898	25.9	289	210	390
			4,304	202	*D*	2	180	356.0	18.8	0.896	26.2	143	185	235
			4,851	67	*D*	2	190	374.0	8.0	0.896	26.2	200	230	330
			4,319	201	*D*	2	185	365.0	9.0	0.896	26.2	140	215	335
			2,460		(*d*)		175	347.0	24.0	0.882	28.7	100	185	215

(*a*) Temperature of 5-per cent distillation, 370 deg. cent (698 deg. fahr.)
(*b*) Insufficient amount of oil.
(*c*) Contained water.
(*d*) Four-week run, with four fuels.

TABLE 6—MISCELLANEOUS DATA ON OIL USED

Pittsburgh Samples													
Kind of Oil	Make of Truck	Truck No.	Number of Miles	Grade of Fuel	Section No.	Initial Point		Percentage Distilled at 300 Deg. Cent. (572 Deg. Fahr.)	Gravity		Viscosity at 100 Deg. Fahr., Saybolt Sec.	Flash-Point, Deg. Fahr.	Fire-Test, Deg. Fahr.
						Deg. Cent.	Deg. Fahr.		Specific	Baume			
New Heavy						300	572.0	......	0.904	24.9	486	430	500
Used Heavy	¾-Ton White	2,114	360	*A*	2	134	273.2	8.0	0.898	25.9	292	180	350
			360	*B*	2	153	307.4	10.0	0.896	26.2	260	170	295
			360	*C*	2	....	.....	10.0	0.897	26.1	200	190	300
			300	*D*	2	165	329.0	4.0	0.902	21.8	333	220	390
		2,106	306	*A*	2	146	294.8	4.5	0.902	25.2	308	205	445
			255	*B*	2	160	320.0	8.0	0.899	25.7	317	190	385
			306	*C*	2	175	347.0	8.0	0.901	25.4	313	210	385
			296	*D*	2	170	338.0	9.5	0.897	26.1	278	115	265
		2,236	210	*A*	2	138	280.4	22.0	0.886	28.0	118	125	160
			180	*B*	2	140	240.0	23.5	0.879	29.3	104	130	185
			190	*C*	2	134	273.2	36.5	0.869	31.1	80	130	165
			170	*D*	2	135	275.0	21.5	0.904	24.9	123	155	200
		2,113	204	*B*	1	144	291.2	9.5	0.897	26.1	256	165	450
			170	*B*	1	130	266.0	9.5	0.896	26.2	261	185	335
			170	*B*	1	140	284.0	7.5	0.898	25.9	284	190	375
			183	*B*	1	132	269.6	10.0	0.895	26.4	246	180	275
		2,105	363	*B*	1	163	325.4	7.0	0.904	24.9	441	255	480
			240	*B*	1	154	309.2	5.0	0.903	25.0	419	250	300
			288	*B*	1	150	302.0	6.5	0.901	25.4	366	230	470
			240	*B*	1	....	.....	6.0	0.902	25.2	390	250	445
		2,705	350	*B*	1	....	.....	12.0	0.891	27.1	214	195	240
			280	*B*	1	....	.....	13.0	0.891	27.1	192	160	220
			341	*B*	1	160	320.0	9.0	0.897	26.1	286	185	310
			393	*B*	1	147	296.6	14.5	0.889	27.5	176	160	200
Used Heavy	2-Ton White	4,868	226	*A*	2	143	289.4	7.5	0.899	25.7	280	175	300
			190	*B*	2	152	305.6	8.5	0.899	25.7	280	185	310
			228	*C*	2	160	320.0	11.0	0.895	26.4	234	180	275
			228	*D*	2	....	.....	15.0	0.891	27.1	186	170	220
		4,878	90	*A*	2	152	305.6	4.5	0.902	25.2	384	230	505
			90	*B*	2	162	323.6	5.0	0.902	25.2	407	170	360
			90	*C*	2	178	352.4	5.5	0.901	26.9	354	225	410
			94	*D*	2	....	.....	6.0	0.899	25.7	338	235	400
		4,876	114	*A*	2	148	298.4	7.0	0.892	26.9	208	180	245
			114	*B*	2	150	302.0	9.0	0.898	25.9	251	190	350
			114	*C*	2	170	338.0	11.5	0.898	25.9	273	170	360
			114	*D*	2	156	312.8	7.0	0.898	25.9	273	220	345
		4,881	246	*B*	1	....	.....	6.5	0.901	26.9	344	220	405
			205	*B*	1	....	.....	4.0	0.903	15.0	436	230	465
			246	*B*	1	160	320.0	6.0	0.901	26.9	367	210	420
			246	*B*	1	167	332.6	3.5	0.903	25.0	453	255	440
		4,867	208	*B*	1	165	329.0	8.5	0.898	25.9	275	185	445
			228	*B*	1	162	323.6	8.5	0.897	26.1	257	180	280
			228	*B*	1	....	.....	6.0	0.899	25.7	310	200	345
			226	*B*	1	163	325.4	9.0	0.896	26.2	262	185	340
		4,873	451	*B*	1	167	332.6	8.5	0.898	25.9	366	180	315
			468	*B*	1	165	329.0	4.5	0.903	25.0	424	235	445
			451	*B*	1	168	334.4	11.0	0.895	26.4	261	185	275
			407	*B*	1	156	312.8	5.5	0.901	26.9	388	210	460
Used Heavy	3-Ton Riker	6,470	170	*A*	2	138	280.4	9.0	0.894	26.6	247	165	350
			188	*B*	2	....	.....	26.0	0.877	29.6	93	180	210
			182	*C*	2	164	327.2	12.0	0.891	27.1	204	170	205
			78	*D*	2	152	305.6	9.5	0.894	26.6	238	175	285
		6,472	108	*A*	2	158	316.4	11.0	0.894	26.6	214	175	245
			81	*B*	2	163	325.4	13.0	0.894	26.6	198	170	215
			97	*C*	2	173	343.4	8.0	0.896	26.2	263	205	400
			114	*D*	2	166	330.8	13.0	0.891	27.1	190	185	225
		6,484	88	*A*	2	145	293.0	10.0	0.893	26.8	209	145	275
			98	*B*	2	....	.....	13.0	0.890	27.3	173	170	195
			90	*C*	2	160	320.0	14.0	0.899	25.7	170	150	215
			88	*D*	2	155	311.0	15.5	0.885	28.2	143	155	195
		6,481	117	*B*	1	150	302.0	12.0	0.891	27.1	190	165	220
			112	*B*	1	143	289.4	20.0	0.883	28.5	142	135	185
			100	*B*	1	127	260.6	12.5	0.891	27.1	185	170	195
			114	*B*	1	(*e*)	.....	18.5	0.882	28.7	128	140	170
		6,485	74	*B*	1	160	320.0	13.0	0.890	27.3	187	190	230
			118	*B*	1	160	320.0	13.5	0.889	27.5	174	160	190
			126	*B*	1	160	320.0	13.0	0.888	27.6	184	170	220
			184	*B*	1	158	316.4	13.5	0.891	27.1	197	120	215
		6,475	212	*B*	1	152	305.6	15.5	0.886	28.0	156	145	195
			199	*B*	1	158	316.4	15.0	0.889	27.5	160	160	185
			202	*B*	1	155	311.0	17.0	0.886	28.0	145	145	170
			169	*B*	1	160	320.0	11.0	0.893	26.8	202	170	250

e)Considerable water.

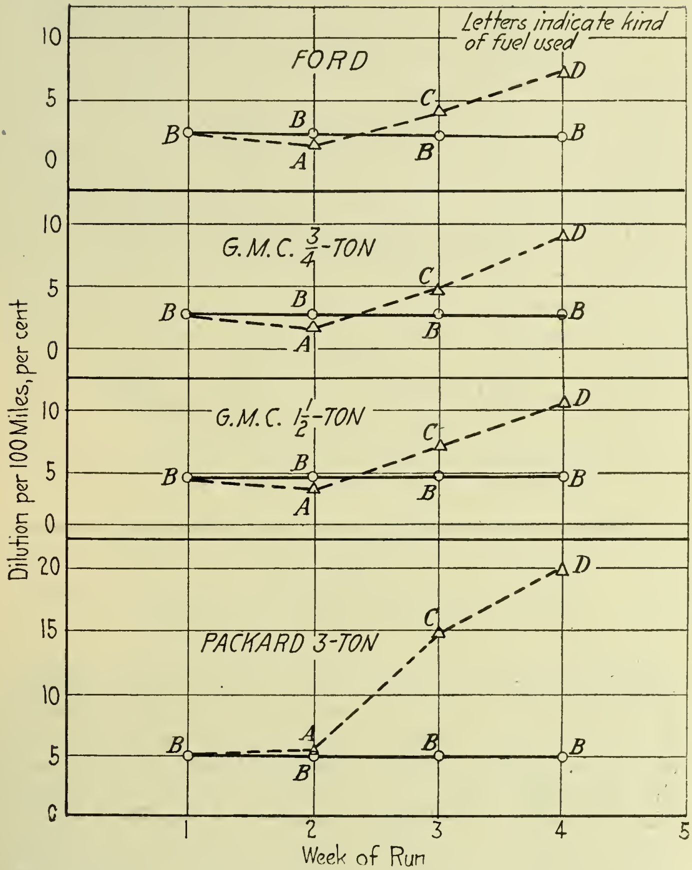

FIG. 5—CHART OF THE CRANKCASE DILUTION IN THE TRUCKS TESTED AT PHILADELPHIA

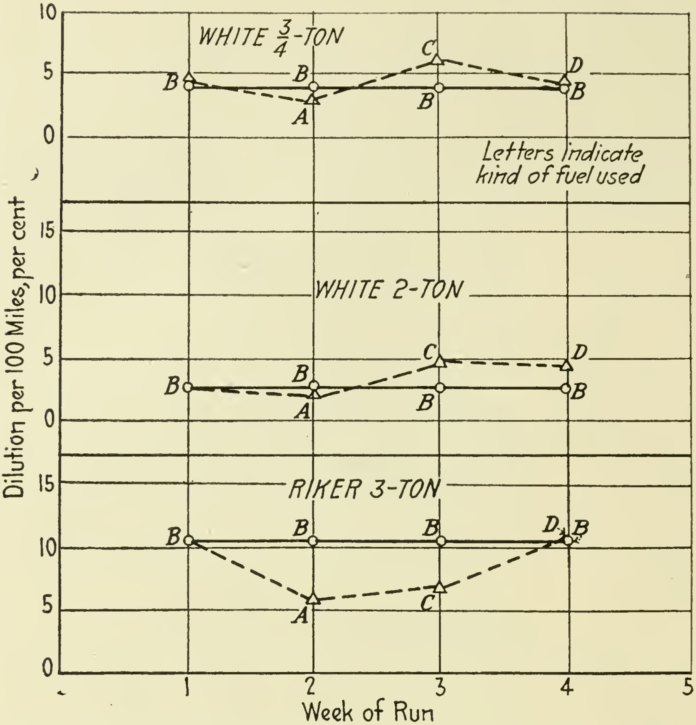

Fig. 6—Chart of the Crankcase Dilution of the Trucks Tested at Pittsburgh

It will be noted that a slightly higher mileage was obtained with the less volatile fuels. This can be explained by the fact that, as the volatility is decreased, the viscosity of the fuel increases and therefore, with a fixed carbureter-setting, slightly less fuel will pass through the carbureter jets and a leaner mixture will result. The difference in mileage therefore is due to a change in the mixture-ratio rather than in the fuel volatility.

The curves in Figs. 2, 3 and 4 give the individual averages of trucks in both cities and the average for both fleets. Curve No. 1 for each make of vehicle shows the actual percentage of change before any correction has been made for the change in the operation on Grade-*B* fuel. Curve No. 2 shows the results after correcting.

A study of these results will show that it seems to make very little difference in the miles per gallon when a truck is operating on any one of the four fuels, under the warm-weather conditions that prevailed while this test was run, and within the range of accuracy possible in a test of this kind. Little or no trouble was experienced in operating on the various fuels, except for a slight difficulty in starting on the Grade-*D* fuel and the necessity for more throttle-opening for proper idling.

METHOD OF EXAMINING OIL SAMPLES

The following procedure was followed when testing used-oil samples for the percentage of dilution. The initial-point of the new oil was ascertained at atmospheric pressure, using a 100-cc. sample in an Engler flask with a water-cooled condensation apparatus. Employing the same apparatus with the used oil, the percentage distilled over at the initial-point of the new oil was noted.

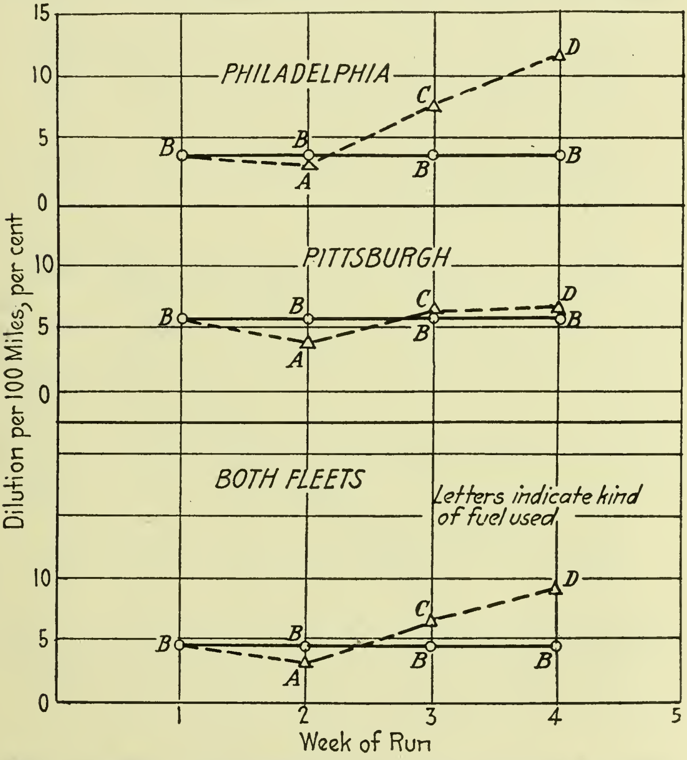

FIG. 7—CHART OF THE AVERAGE CRANKCASE DILUTION AT PHILADELPHIA AND AT PITTSBURGH AND THE GENERAL AVERAGE FOR BOTH FLEETS

This was termed the percentage of dilution. To check this method, samples of new oil were diluted with known percentages of the distillates from the used oil and distillations run on these samples. The results obtained agreed with the known percentage of dilution.

Following the method outlined, the results obtained for the percentage of dilution from the several trucks operating on the different fuels are shown in Tables 5 and 6. Used-oil samples were also examined for changes in viscosity, specific gravity, flash-point and fire-test; the results are shown in Tables 5 and 6 also. The curves in Figs. 5 and 6 show relative crankcase dilution per 100 miles for the four fuels. The curves in Fig. 7 give the fleet averages for both Philadelphia and Pittsburgh. In considering the average dilution for both fleets, it will be noted that, compared with the trucks

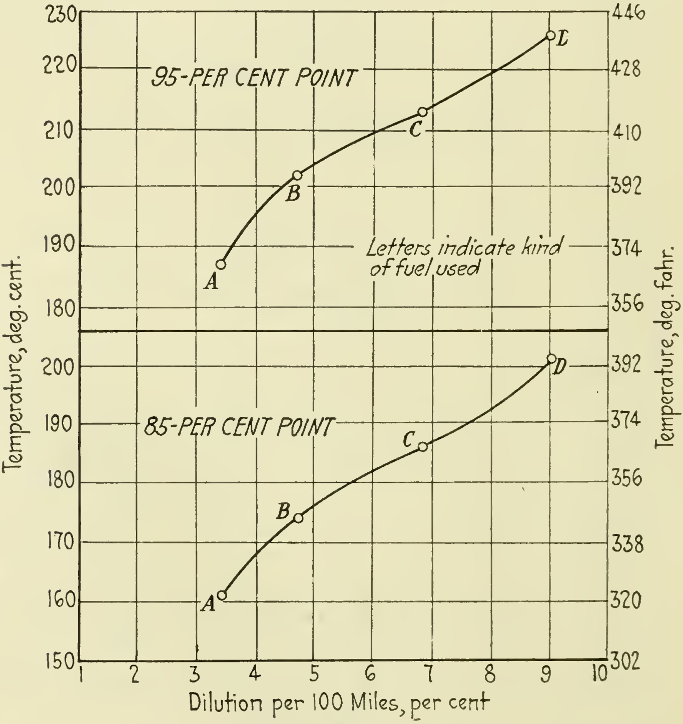

Fig. 8—Curves Showing the Relation between the Percentage of Dilution per 100 Miles and the Temperatures of the 85 and 95-Per Cent Distillation Points

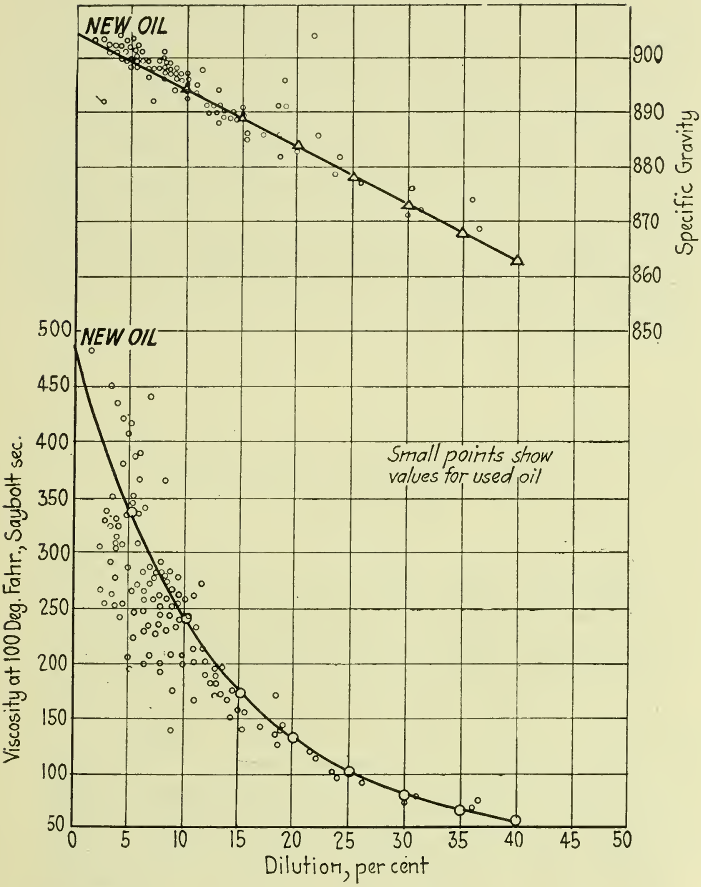

FIG. 9—CURVES SHOWING HOW THE VISCOSITY AND THE SPECIFIC GRAVITY OF THE NEW LUBRICATING OIL CHANGED WHEN DILUTED WITH VARYING PERCENTAGES OF DISTILLATES FROM THE USED OIL

run on Grade-*B* fuel, those run on Grade-*A* fuel had less crankcase-oil dilution, and that Grades *C* and *D* gave correspondingly more crankcase-oil dilution. From these values the curves shown in Fig. 8 were plotted for the percentage of dilution against the temperature from the distillation curves of the 85 and 95-per cent points. The curves show that, as the end-point of the fuel is increased, the dilution of the oil is materially increased.

To compare the change in viscosity and specific gravity due to dilution, samples of new oil were diluted with the distillates from used oils to the following percentages: 5, 10, 15, 20, 25, 30, 35 and 40 per cent. These samples were then examined for viscosity and specific gravity. The results are shown in the curves of Fig. 9. The points in the small circles represent the changes noted in the viscosity and the specific gravity of used oils of known percentage of dilution.

OIL CONSUMPTION[1]

By A A Bull[2]

The object of the paper is to consider some of the fundamental factors that affect oil consumption; it does not dwell upon the differences between lubricating systems. Beyond the fact that different oils apparently affect the oil consumption and that there is a definite relation between viscosity and oil consumption, the effect of the physical characteristics, or the quality of the oil, does not receive particular attention.

The methods of testing are described and the subject is divided into (*a*) the controlling influence of the pistons, rings and cylinders; (*b*) the controlling influence of the source from which the oil is delivered to the cylinder wall. The subject is treated under headings that include the piston-ring; the effects of oil-return holes, side-clearance and ring motion; thin rings; influence of piston fit; efficiency of the scraper-ring; ring and cylinder contact; carbonization and spark-plug fouling; oil-supply control; influence of oil viscosity; effects of dilution; external oil leaks and breather discharge, and the influence of controlling lubrication in proportion to throttle opening.

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[1] Semi-Annual Meeting paper.

[2] M.S.A.E.—Chief engineer, Northway Motors & Mfg. Co., Detroit.

The subject of oil consumption, relating particularly to automotive engines, is one of importance. I realize the attention and effort that have been directed toward a solution of many of its problems by both the oil companies and the engine and vehicle builders.

From the owner's standpoint, the question of oil consumption in all its phases becomes of interest sooner or later. He is perhaps more interested in the troubles that he has experienced that result from excessive oil consumption or, as it is generally understood, oil-pumping. Over-oiling of this character, which occurs under average driving conditions and causes excessive carbon deposit and fouling of the spark-plugs, results in unsatisfactory operation of the engine. However, these symptoms are not necessarily characteristic, and engines that are not directly subjected to these troubles will use excessive quantities of oil; from an economic standpoint, this phase warrants consideration. The economic viewpoint of oil consumption is more important in truck and tractor engines, or those that are used in commercial service, where oil consumption affects operating costs. The object of this paper is to consider some of the fundamental factors that affect oil consumption and it will not dwell extensively on the difference in lubricating systems which, as will be self-evident, have an influence on oil consumption. Different oils apparently affect oil consumption and there is a definite relation between viscosity and oil consumption which can be attributed to several causes that will be discussed but, beyond this, the effect of physical characteristics or quality of the oil will not receive particular attention.

The extent of dilution has an important bearing on oil consumption in its effect in lowering the viscosity of the oil so that it cannot be controlled so readily. In passenger-car engines particularly the operator is likely to believe that his oil consumption is very small for the reason that the added diluent compensates for the actual oil consumed.

Apart from over-oiling as evidenced by excessive carbon, fouled spark-plugs and the like, what constitutes reasonable oil consumption? In commercial service, where operating costs must be kept at the minimum, perhaps the oil consumption should bear a definite relation to fuel consumption. The actual amount of oil required to lubricate the engine properly, particularly the pistons and cylinders, is surprisingly small and, because the oil pumped to the combustion-chamber is the chief source of oil consumption, attention necessarily is directed to controlling the extent to which oil reaches the combustion-chamber. The wide variation in oil consumption with different heavy-duty engines is well represented by the figures in Table 1. These figures have been collected from engines of different types, working, in most cases, under actual service conditions. The basis of comparison is the oil consumed per brake-horsepower-hour and, while this is not a direct indication of the relative consumption, it is based on the assumption that the engine speeds are nearly the same and that the horsepower output therefore indicates the size of the engine.

Oil consumption in passenger cars usually is computed on the basis of miles per gallon. It is possible to obtain figures as low as 50 and as high as 2000 miles per gal. As indicated previously, in ordinary service with low average speeds, the oil consumed may be negligible; in fact, it is possible that more fluid can be taken from the crankcase after service than was originally put in. On the other hand, engines or cars that will exhibit this characteristic will consume unreasonable quantities at continued speeds of 30 m.p.h. or over. An instance of this character which indicates a fairly general condition is indicated in Table 2.

TABLE 1—OIL CONSUMPTION

Engine	Type	Speed, r.p.m.	Horsepower
A	Tractor	....	....
A	Tractor	....	....
A	Tractor	....	....
A	Tractor	....	....
S	Tractor	....	20.0
F	Tractor	....	18.0
WB	Tractor	....	25.0
C	Tractor	....	18.0
C	Tractor	....	27.0
AR	Tractor	....	30.0
AR	Tractor	....	20.0
RI	Tractor	....	16.0
I	Tractor	....	20.0
I	Tractor	....	16.0
N_1	Truck[a]	....	37.0
N_2	Truck[a]	....	30.0
N_3	Truck[a]	....	27.5
A	Truck	1,500	33.0
A	Truck	1,200	35.0

[a] Maximum dilution, 1.5 to 2.0 per cent.
[b] Continuous.

TABLE 2—OIL CONSUMPTION

Car No.	1	2	1[c]	2[c]
Number of Cylinders	8	8	8	8
Average Speed, m.p.h.	15	15	40	40
Total Miles Run	600	500	50	50
Oil Consumption[d]	0	0.5	4-5.75	11
Dilution per Gallon, per cent	40	10	0	0
Oil Consumption, Excluding Diluent, miles per gal.	860	835	92	581

[c] Same cars on a speedway, at 40 m.p.h.

[d] The consumption is given in gallons in the first and second columns and in ounces in the third and fourth columns.

METHODS OF TESTING

In the study of oil consumption there are several methods in which comparative tests can be conducted. First, of course, tests under actual operating conditions are necessary to determine the average oil consumption and the general efficiency of the lubricating system. However, this method is not satisfactory in studying fundamentals, the chief difficulty being that it is not possible to have or control similar conditions of operation and equipment. Several methods were used in making observations in the laboratory.

(1) Obtaining the oil consumption under working conditions, properly controlling all the affecting factors

TABLE 1—OIL CONSUMPTION (*Concluded*)

Hours	Oil Consumption, gal.		
Operated	Total	Per Hr.	Per B. Hp-Hr.
97.00	......	0.2070	......
379.57	......	0.1430	......
426.59	......	0.1370	......
2,997.00	......	0.1500	......
37.00	9.7500	0.2630	0.01320
34.00	3.7500	0.1100	0.00620
44.00	3.3800	0.0770	0.00370
40.00	8.0000	0.2000	0.01110
37.00	11.5000	0.3100	0.01150
35.00	15.5000	0.4420	0.01470
44.00	5.0000	0.1120	0.00560
33.00	4.3800	0.1330	0.00830
30.00	8.0000	0.2670	0.01330
32.00	4.7500	0.1490	0.00930
5.00	0.4250	0.0850	0.00230
4.00	0.0935	0.0234	0.00580
4.00	......	......	......
1,000.00[b]	......	0.1200	0.00364
500.00[b]	......	0.0500	0.00143

(2) Measuring the quantity of oil pumped through the exhaust of individual cylinders

(3) Studying the extent of the oil passing the pistons to the combustion-chamber with the cylinder-head removed

Method (3) gives very good results when checking fundamentals and is related closely to the results obtained by methods (1) and (2). In any case, in conducting oil-consumption tests it will be found that considerable variation in results is likely to be obtained unless great care is taken and until some fundamental has really been found. For this reason any small improvements or changes in oil economy that may be effected by the use of or changes in detail are not to be relied upon.

In studying the extent of oil-pumping by an observation of the amount collecting on top of the piston, the quantity of oil passed by is in excess of that which is used under working conditions, which assumption is drawn from the relative quantities of oil used. Every stroke of the piston functions the same, not being affected by the pressures existing in the cylinder during the cycle, the compression, expansion and exhaust pressures evidently having a beneficial influence and exercising more control than the effects of suction.

Before proceeding to a complete analysis, it is necessary to separate the subject into two classes (*a*) the controlling influence of the pistons, rings and cylinders; and (*b*) the controlling influence of the source from which the oil is delivered to the cylinder wall.

It generally is assumed that the piston-rings exercise the most important part in controlling the extent of oil-pumping. This is substantiated by the extraordinarily large number of different types of ring that claim in one way or another to control the oil consumption and eliminate the troubles usually associated with over-oiling. There are well established definite and fundamental principles that must be observed in the application of the rings to the piston, and the problem is one of maintaining the efficiency indefinitely. Unfortunately, the majority of replacement rings have no fundamental quality in themselves that affects or exercises control over the oil consumption and, in most instances where replacements are made, equally good results would be obtained by employing ordinary plain piston-rings.

An engine upon which some of the tests were conducted is illustrated in Fig. 1; its construction is obvious.

It has a full pressure-feed lubricating system and separate cylinder sleeves, and the oil reservoir is divided into several compartments to filter the oil adequately.

THE PISTON-RING

While the piston-ring primarily is used to retain compression in the cylinder, it has become necessary that the ring regulate or control the extent of the lubrication of the cylinder walls. Let us consider for a moment the action of the piston-ring in traveling the surface of the cylinder already coated with oil-film. During the expansion and suction strokes, when the ring is traveling down, it is presumed that it scrapes or pushes the excess oil adhering to the cylinder walls in front of it. In any case, this is exactly what the ring is required to do. The factors that apparently affect the efficiency of the ring in performing this duty are the thickness of the oil-film and the volume of oil, the unit pressure between the ring and cylinder, which determines its ability to break-down the oil-film, and the character of the leading edge of the ring.

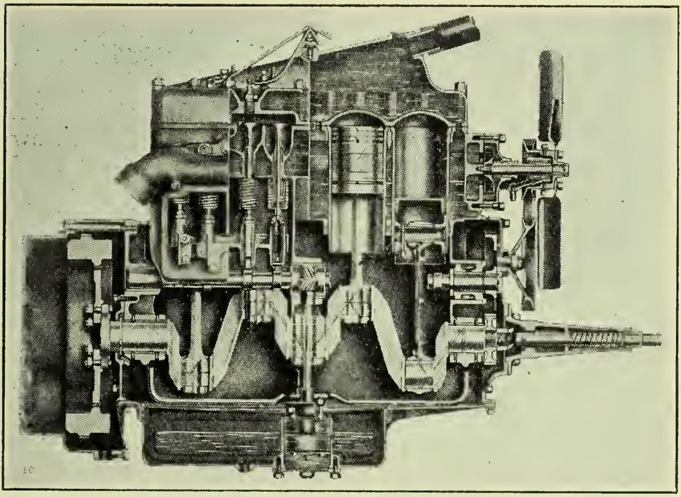

FIG. 1—SECTIONAL VIEW SHOWING THE CONSTRUCTION OF THE ENGINE UPON WHICH THE TESTS WERE MADE

The thickness of the oil-film depends upon the viscosity. In this connection it should be emphasized that the temperature of the oil in the main body of the oil-pan is not necessarily at the same temperature as the oil-film on the

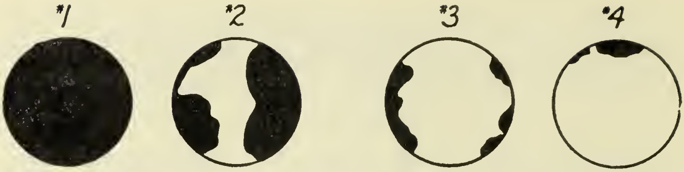

Fig. 2—Test Results Obtained at a Speed of 500 R.P.M., Oil and Water Temperatures of 109 and 127 Deg. Fahr. Respectively and an Oil Pressure of 12 Lb.

Cylinders Nos. 1 and 2 Were Equipped with a Single Piston-Ring and Had No Return Holes; the Other Two Cylinders Had Standard Three-Ring Equipment

cylinder wall and, except under conditions where the cylinders are relatively cool, there is little variation in the viscosity on the cylinder wall. Quantitatively, of course, there will be more or less oil on the cylinder wall, depending upon the extent to which the oil is thrown from the connecting-rod and crankshaft bearing which, as succeeding analysis shows, exercises a very decided influence.

The unit pressure between the ring and the cylinder wall has some effect on the efficiency of the ring in displacing the oil from the cylinder wall. With conventional cast-iron piston-rings, the wall pressure necessarily is limited by the stress that can be imposed on the ring material if it is to operate against the cylinder wall due to its inherent elasticity. The tension that can be obtained is adequate, although the increase in the unit pressure, obtained by a relief or a groove in the face of the ring or by the use of thin rings with a supplementary spring, will improve the scraping efficiency. As succeeding tests will indicate, however, this feature in itself is not sufficient to give the proper results. The character of the leading edge of the ring also has its effect and, unless it is relatively sharp, the ring will ride over the surface of the oil-film. For this reason a ring with a bevel is of

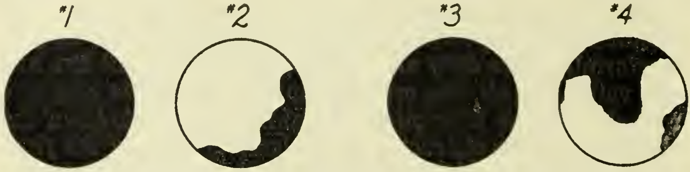

Fig. 3—Test Results Obtained with the Same Equipment as in Fig. 2 at a Speed of 1000 R.P.M.

The Difference in the Oil Deposits in Cylinders Nos. 2 and 3 as Compared with Fig. 2 Should Be Noticed benefit and, on the up-stroke, the ring will ride readily over the film. Along these lines, it is believed that the type of ring having an inherent twist, so that the unit pressure on the bottom edge is in excess of that on the top, may be beneficial.

EFFECT OF OIL-RETURN HOLES

Substantial force is required to displace the oil-film adjacent to the edge of the ring, and it is evident that a high pressure exists which, unless suitably relieved by the presence of oil-return holes, breaks-down the seal between the face of the ring and cylinder. Reference to Figs. 2, 3 and 4 clearly illustrates the effect of oil-return holes. Fig. 5 shows the same ring equipment operating at the normal oil temperature of 142 deg. fahr., but with oil-return groove and holes immediately below the ring.

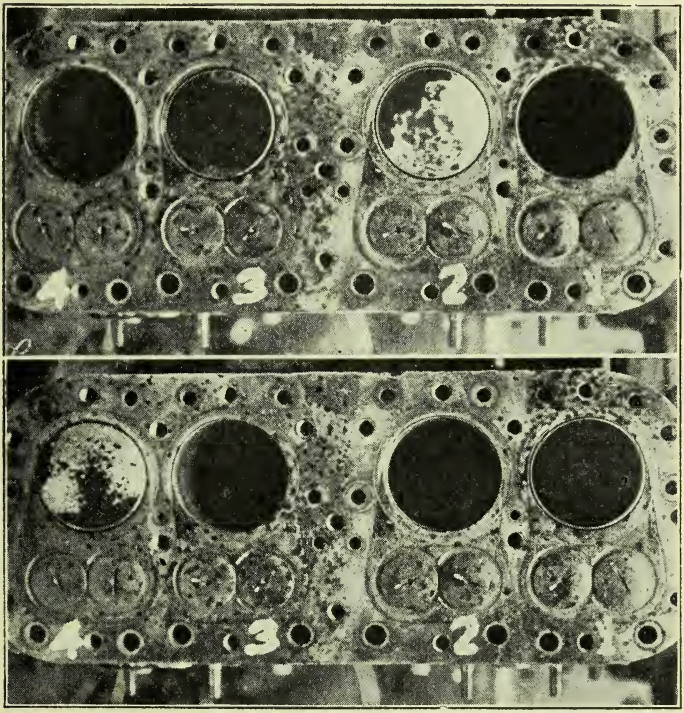

FIG. 4—PHOTOGRAPHS OF THE OIL DEPOSITS SHOWN (ABOVE) IN FIG. 2 AND (BELOW) IN FIG. 3

The improvement is evident. The ring used in the preceding test had a good fit in the groove. However, it is recognized that sidewise clearance increases with use.

The diagrams and photographs that refer to the test results are presented as representative selections of a considerable number of tests. Both photographs and scale diagrams are shown in some instances. In some cases, scale diagrams alone are shown because they offer a better comparison. The variation in ring construction is made on cylinders Nos. 1 and 2, the same rings being used in each case and being placed in the same circumferential position. Cylinders Nos. 3 and 4 have standard three-ring equipment, except where otherwise noted. Cylinders Nos. 1 and 3 have similar oil distribution from the crankshaft, while cylinders Nos. 2 and 4 differ as will be explained later. The relative effect of change in ring construction on the amount of oil-pumping with the different oil distribution can be compared in cylinders Nos. 1 and 2.

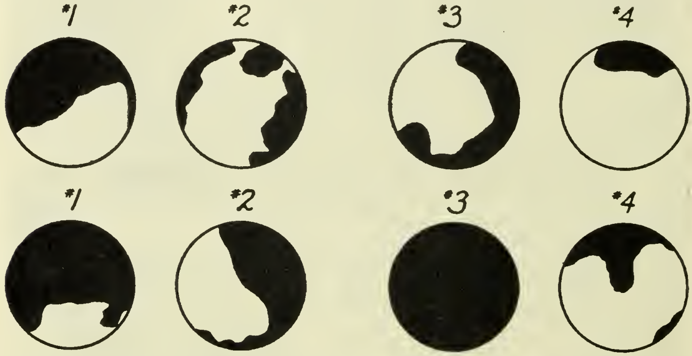

Fig. 5—Test Results Obtained (Above) at 500 R.P.M. and (Below) at 1000 R.P.M.

Cylinder No. 1 Had One Ring with Oil Holes Immediately below the Ring

In Fig. 2 cylinders Nos. 1 and 2 were equipped with one ring and had no oil-return holes. Cylinders Nos. 3 and 4 were provided with standard three-ring equipment. The oil pressure was 12 lb. per sq. in.; the oil temperature, 109 deg. fahr.; the water temperature, 127 deg. fahr.; the speed, 500 r.p.m.; and the duration of the test, 3 min.

In Fig. 3 the same conditions prevailed as in Fig. 2, except that the speed was 1000 r.p.m. and the duration

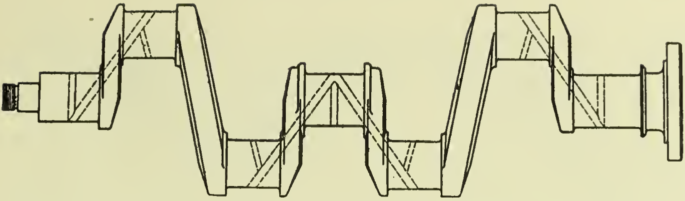

FIG. 6—CRANKSHAFT IN WHICH OIL-HOLE LOCATION COULD BE CHANGED FOR THE EXPERIMENTS

of the test was 1 min. The difference in the increase in the oil deposit on cylinders Nos. 2 and 3 should be noted.

In the upper portion of Fig. 5 the oil pressure was 12 lb. per sq. in.; the oil temperature, 156 deg. fahr.; the speed, 500 r.p.m.; and the duration of the test, 3 min. Cylinder No. 1 had one ring, with oil-return holes immediately below the ring. In the lower portion of Fig. 5 the same conditions prevailed as in the upper portion, except that the speed was 1000 r.p.m. and the duration of the test was 1 min.

Fig. 6 shows the crankshaft, in which the oil-hole location could be changed during the experiments.

Fig. 7 illustrates part of the equipment used for measuring the quantity of oil pumped through the exhaust.

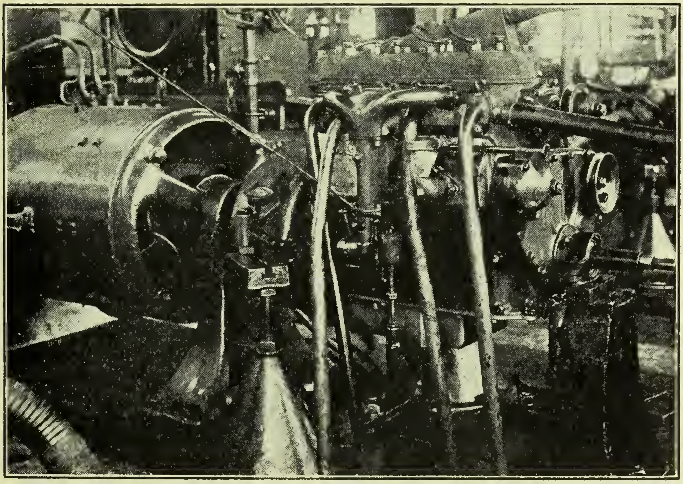

FIG. 7—PART OF THE EQUIPMENT FOR MEASURING THE QUANTITY OF OIL PUMPED THROUGH THE EXHAUST

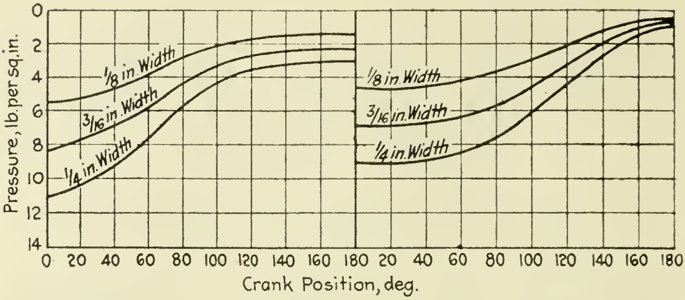

FIG. 8—DIAGRAM SHOWING THE UNIT PRESSURE ON THE RING-GROOVE PRODUCED BY FRICTION AND INERTIA FORCES OF RINGS OF DIFFERENT WIDTHS AT 1000 R.P.M.

EFFECTS OF SIDE-CLEARANCE AND RING MOTION

Digressing for a moment, let us consider the effects of side-clearance on the ring. During the down-stroke of the piston, the ring is contacting with the upper face of the groove leaving the clearance at the lower edge; on the up-stroke, the condition is reversed. Having recognized that oil pressure is built up on the under side of the ring, it follows that, if the ring is not seating on the groove, the oil will pass through to the rear of the ring.

The three factors that control the movement of the ring relative to the piston are the

(1) Inertia of the ring itself or its resistance to acceleration
(2) Friction of the ring against the cylinder wall
(3) Effects of pressure within the combustion chamber

Fig. 8 shows the inertia and friction forces operating on the ring at 1000 r.p.m., indicating that the position of the ring does not change in respect to the groove. At higher speeds, however, the position changes before the end of the stroke.

The effect of side-clearance on oil-pumping is illustrated in the two portions of Fig. 9. The oil is pumped up very rapidly, with or without an oil-return hole below the ring. At 500 r.p.m. piston No. 1 is covered completely in 1 min., but it becomes full in 0.3 min. at a speed of 1000 r.p.m. In the upper portion of Fig. 9 cylinders Nos. 1 and 2 each have one ring with a 0.005-in. side-clearance and cylinders Nos. 3 and 4 are provided with standard three-ring equipment. At a speed of 500 r.p.m. and a duration of test of 3 min., cylinder No. 1 fills up in 1 min. In the lower portion of Fig. 9 the conditions are the same as in the upper portion, except that the speed is 1000 r.p.m. and the duration of test 1 min. Cylinder No. 1 fills up in 0.3 min. Fig. 9 should be compared with Fig. 5.

In this connection consideration should be given to the extent of the side-clearance that occurs in the ring-groove. With cast-iron pistons and rings the initial fit can be established fairly well and, under operating conditions, the expansion of the ring is practically the same as that of the groove; consequently, no additional clearance occurs until the ring or groove becomes warm. It is not possible to determine when increased clearance will occur, as this depends on the conditions under which the engine is operating. With a three-ring piston the actual clearance will vary from the top down in about the following relation: upper, 0.010 in.; second, 0.006 in.; and third, 0.003 in.

The extent of side-clearance unquestionably is the factor that is responsible for any difference between cast-iron and aluminum pistons. Regardless of the accuracy of the fit of the ring in the groove, the difference in expansion between the ring-groove and the ring on an aluminum piston produces initial clearance in excess of that required to effect the proper seal and increase in the rate of wear between the ring and groove because of the initial reciprocation. The use of oil-return holes in the ring-groove at the back of the ring is very bene-

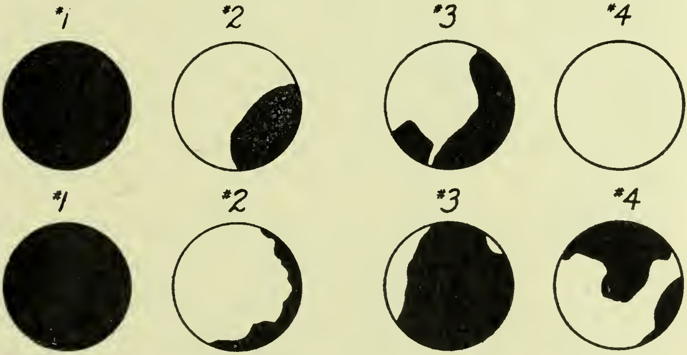

Fig. 9—Test Results Obtained (Above) at 500 R.P.M. and (Below) at 1000 R.P.M.

Cylinders Nos. 1 and 2 Had One Ring with a Side Clearance of 0.005 In., While the Other Two Had Standard Three-Ring Equipment. At the Lower Speed Cylinder No. 1 Filled in 1 Min. and in 0.3 Min. at the Higher Speed. These Diagrams Should Be Compared with Those in Fig. 5

ficial when the ring has side-clearance so that the oil has access to the space back of the ring.

The two halves of Fig. 10 indicate the benefits of these oil-return holes. In the upper view the engine is operated at 500 r.p.m. and at 1000 r.p.m. in the lower, the equipment being the same. It is obvious that, with holes so located, particularly when used in a ring-groove above the piston-pin, the ring is useless for retaining the compression. Cylinders Nos. 1 and 2 each have one ring with 0.005-in. side-clearance and oil-return holes in the ring-groove. Cylinder No. 3 has rings ⅛ in. wide and no relief-holes. The duration of the test was 3 min. for the upper view and 1 min. for the lower. Fig. 10 should be compared with Fig. 9.

THIN RINGS

The influence of ring thickness on side-clearance has received considerable attention, particularly in the case of aluminum pistons. Reduction in ring thickness unquestionably reduces the relative clearance. The weight of the ring itself is reduced, consequently decreasing the inertia effects of the ring. The relative forces with rings of different thicknesses are represented in Fig. 8. Fundamentally, however, they cannot solve the difficulty. A multiplicity of thin rings in a narrow groove would provide a good solution. They must be made of steel rather than cast-iron, and they must be hard so that they will not lap the bore and cause excessive wear. An effective unit-pressure of the ring against the wall could be well maintained and, likewise, the inertia effects of the portions of the ring would be dampened out by the oil-film between the several rings. Circumferential sealing also would be improved. Piston No. 3 in Fig. 10 is equipped with four ⅛-in. rings above the piston-pin without any relief-holes, from which it would seem that no benefits are obtained.

Combining all the essential requirements of a ring, it seems that a ring of two-piece construction capable of sidewise expansion is desirable. It is important, however, that sufficient wall pressure be provided; otherwise the benefits of the side-seal are lost with the inability of the ring to displace the oil from the cylinder wall properly.

The clearance of the piston in the cylinder unquestionably has some effect on the functioning of the ring; however, this is not because an increased clearance between the piston and cylinder permits a larger quantity of oil to accumulate. From a fundamental standpoint the oil-film is not likely to be any thicker with a loose piston than a tight one. The oil pressure built up ahead of the ring, however, is increased with the closer fit because the oil-film is confined more closely.

Piston fit in the cylinder has an appreciable effect on the wear of the rings; on both the outside, which contacts with the cylinder wall, and the sides contacting with the ring-groove. It is evident that the piston will rock in the cylinder to the extent of the clearance, and that it tends to form a convex surface on the outside of the ring, completely destroying its oil-scraping efficiency. The continual sliding back-and-forth of the ring upon the seat wears the ring-groove very rapidly, and it is believed it is more responsible for wear than the actual pressure exerted by the ring on the groove because of its reciprocation.

On the basis that the ordinary ring is likely to carry up as much oil as it will scrape down, the use of the fourth ring below the piston-pin is of little advantage, except that it maintains its fit and condition very much better than the others and therefore has merit. Fundamentally, the use of a fourth ring placed at the lower end of the piston increases the amount of oil to be handled. It follows that the scraper-ring itself travels over a film surface equivalent to the stroke of the

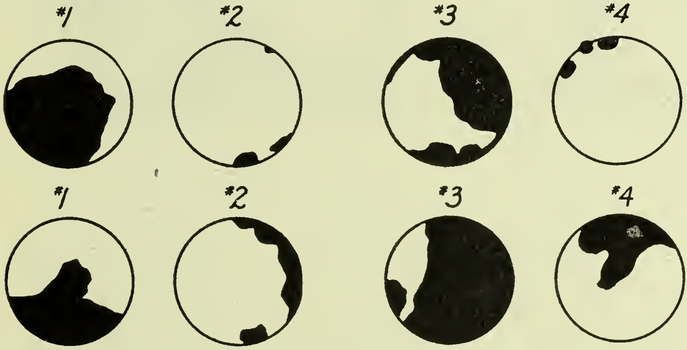

Fig. 10—How the Presence of Oil-Return Holes Affects the Formation of Oil Deposits

Cylinders Nos. 1 and 2 Were Equipped with One Ring Having a Side Clearance of 0.005 In. and Had Oil-Return Holes in the Ring-Groove, While Cylinder No. 3 Was Fitted with Rings ⅛ In. Wide and No Relief Holes. The Upper Drawing Shows the Results Obtained at 500 R.P.M. and the Lower One Those Obtained at 1000 R.P.M. These Results Should Be Compared with Those of Fig. 9

piston, while the upper rings also travel over a surface of similar length, the accumulated travel being greater and the oil consequently being relayed from one ring to the other. It is interesting to observe that, without any ring on the piston, absolutely no oil is passed up, the piston sliding over the lubricating film without disturbing it.

RING AND CYLINDER CONTACT EFFECTS

Plain rings change their position circumferentially in the cylinder and, if the cylinders are out-of-round, which is usual, due to unequal expansion and the like, considerable oil will be passed by the rings. It was evident in all the tests that have been conducted that the location of the ring circumferentially is very important and, in duplicating results, it is necessary to pin the rings so that they cannot rotate. To illustrate the extent to which contact between the ring and cylinder affects consumption, the following comparison is submitted.

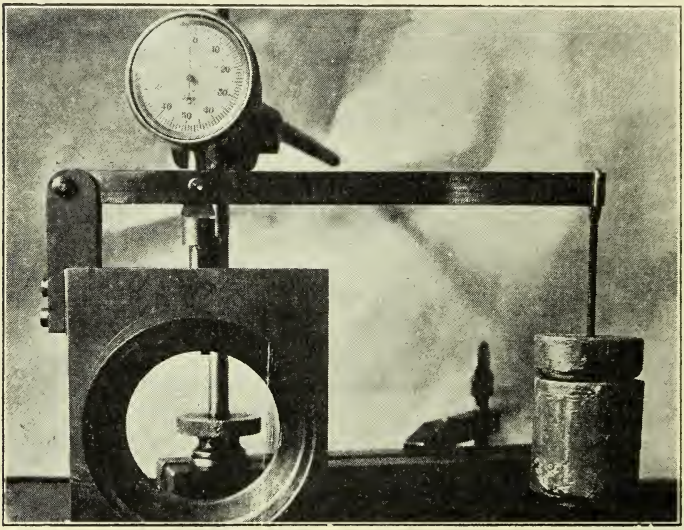

FIG. 11—APPARATUS USED FOR THE DETERMINATION OF PISTON-RING PRESSURE

With Multiple-Piece Rings, the Pressures Were Taken with the Ring Installed in the Groove

As a result of using separate cylinder sleeves, it was possible to obtain an ideal comparison. The pistons and rings remained undisturbed and new cylinder sleeves

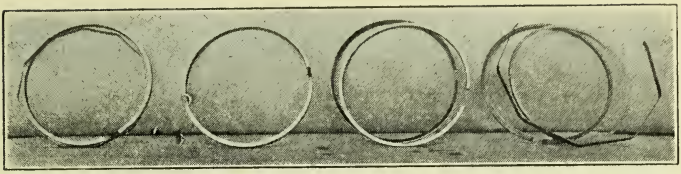

Fig. 12—Some of the Various Types of Ring with Which the Experiments Were Conducted

were used, the engine having seen considerable service. With old rings and cylinders, the oil consumption was 1 gal. per 400 miles and with the same rings and pistons, but with new cylinders, the oil consumption was 1 gal. per 150 miles.

Summarizing the fundamental factors of the piston ring, we conclude that

(1) Drain-holes are absolutely essential
(2) An angular-faced piston-ring is beneficial
(3) The proper mechanical fit between the ring and the groove is essential
(4) Oil relief-holes in the rear of the groove are beneficial when side-clearance occurs
(5) The slot in the ring is unimportant
(6) The number of rings is unimportant

Fig. 11 shows the apparatus used for the determina-

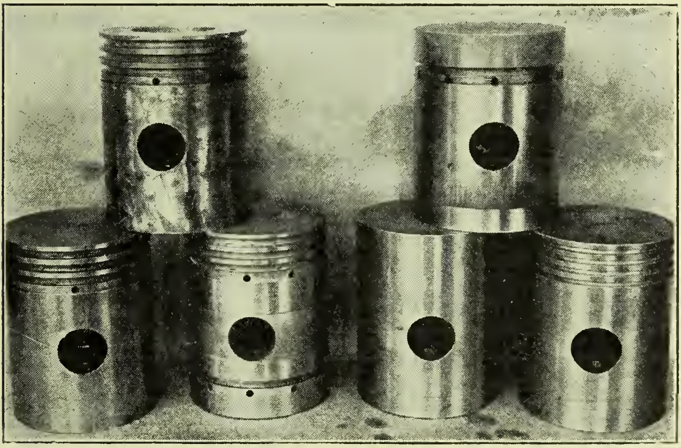

Fig. 13—Several Types of Piston Installed During the Tests

Pistons Having One Ring-Groove Only Were Employed in Obtaining the Data on Ring Characteristics tion of piston-ring pressure. With multiple-piece rings, the pressures were taken while the ring was installed in the groove.

Fig. 12 shows some of the various types of ring with which the experiments were conducted, and Fig. 13 illustrates several types of piston that were installed during the tests. Pistons having one ring-groove only were employed in obtaining the data on ring characteristics.

Table 3 gives piston-ring pressures taken in the groove and taken free. Table 4 shows the relative unit wall-pressure with change in contact area and face width.

Passing from a consideration of the mechanical aspects of rings and pistons in the controlling of oil passing to the combustion-chamber, mention should perhaps be made of some of the factors affecting carbonization and the fouling of spark-plugs after the oil has reached the combustion-chamber. First, of course, is the nature of the lubricant. It usually is considered that the heavier oils produce more carbon deposit, although this is not always true. The extent of the free carbon in the lubricant has little effect upon combustion-chamber deposits. In this respect the benefits of proper heat conductivity on the piston-head, to prevent excessive carbon deposit

TABLE 3—PISTON-RING PRESSURES

Ring Diameter, Outside, 3½ in.; Ring Width, $\frac{3}{16}$ in.

Kind of Ring	Weight in Pounds, at Deflection of		
	0.001 in.	0.002 in.	0.003 in.
Weight Taken on Ring Not Installed in Groove			
Concentric, Hammered	4.96	6.10	6.650
Two-Piece, Outer	1.28	1.52	1.850
Two-Piece, Inner	.....	0.64	0.775
Three-Piece, Spring Type, Outer	0.00	0.00	0.000
Three-Piece, Spring Type, Inner	0.00	0.52	0.640
Weight Taken with Rings Installed in Groove			
Hammered	5.20	6.40	7.000
Two-Piece	3.00	3.65	4.400
Three-Piece, Spring Type	6.40	10.40	13.200
Hammered, Spring Type	15.90	24.52	33.000

TABLE 4—RELATIVE WALL PRESSURE OF RING OF 3½-IN. OUTSIDE DIAMETER

Outside Diameter of Ring, 3½ in.			Deflection		
			.....	0.001 in.	0.001 in.
Ring Width, In.	Normal Ring Tension, Lb.	Developed Contact Area, Sq. In.	Wall Pressure, Lb. per Sq. In.		
			Normal	Tight	Loose
$\frac{3}{16}$	4.5	0.665	6.75	8.25	5.25
⅛	4.5	0.404	11.20	13.60	8.65
$\frac{3}{32}$	4.5	0.341	13.20	16.20	10.20

on the inside, should be observed. The location of the spark-plug is of great importance, as in many cases, of course, frequent fouling of the spark-plugs has been a result of improper location, the least evidence of over-oiling immediately affecting its operation.

A typical condition of cylinder-head carbon-deposit is shown in Fig. 14. The maximum deposits are on the side of the cylinder upon which most of the oil is thrown from the crank, the spark-plugs being sufficiently removed from this zone to prevent their becoming carbonized.

Control of the Oil Supply

Control of the oil supply, particularly that from the bearings to the cylinders, unquestionably has more influence on the oil consumption than the mechanical condition of the pistons and the rings. Stated briefly, it is very evident that, under the best conditions, the rings and the pistons can exercise only a very definite control over the amount pumped to the combustion-chamber.

In the discussion of the mechanical aspect of the pistons and the rings, it is concluded that there are definite fundamental conditions that must exist in order to control best the oil passing to the combustion-chamber, and the problem of the piston-rings and the pistons is to maintain these fundamental conditions. This same argument holds for the control from the source of supply. It is probable that, with the very close fits of bearings, the extent of oil discharge to the cylinders will be of such magnitude that the rings can control it properly. As wear of the bearings occurs, the discharge is increased greatly and this is true particularly where an oil-regulating mechanism is provided that will maintain the oil pressure even though the bearings become loose.

It will be further evident from the analysis yet to be made that, in most instances, the effect of higher speed is to increase the quantitative discharge from the bearings and it will be shown that, while ordinarily higher speed will increase the oil consumption considerably, it is believed that this is not due to any change in functioning of the piston-rings in keeping the oil from passing into the combustion-chamber. In other words, the rings will be almost equally efficient within the ranges of piston speed under average conditions, if they are fitted properly.

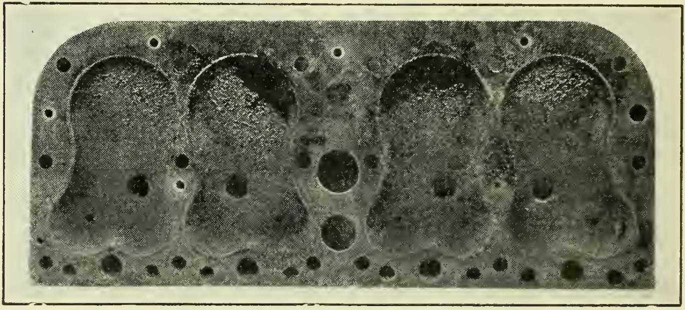

FIG. 14—A TYPICAL CYLINDER-HEAD SHOWING THE CARBON DEPOSIT IN A TRACTOR ENGINE

Trucks and tractor engines that operate a large percentage of the time at governed speeds are affected greatly in their oil consumption by the bearing condition. It is also true of passenger-car engines that a car will show a reasonable economy when operated at ordinary driving speeds and, while this economy may be more or less superficial as a result of the dilution that takes place, the same car running at higher speeds, where dilution does not become a factor, will consume unreasonable quantities.

The consideration of the oil discharge from the crankshaft is, naturally, confined largely to pressure-feed systems. With splash feed for the connecting-rods, the amount thrown onto the cylinder wall depends, of course, on the dip. It is not controlled as easily except through

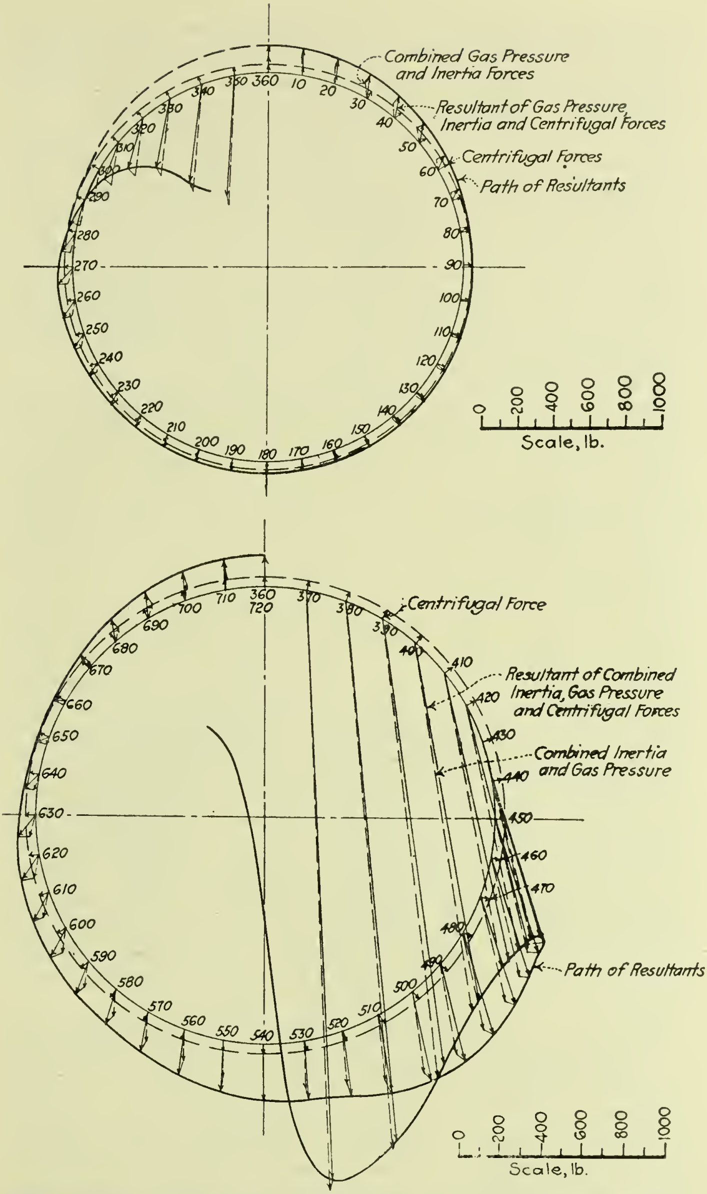
Combined Gas Pressure and Inertia Forces
Resultant of Gas Pressure, Inertia and Centrifugal Forces
Centrifugal Forces
Path of Resultants
0 200 400 600 800 1000
Scale, lb.
Centrifugal Force
Resultant of Combined Inertia, Gas Pressure and Centrifugal Forces
Combined Inertia and Gas Pressure
Path of Resultants
0 200 400 600 800 1000
Scale, lb.

FIG. 15—CRANKPIN BEARING PRESSURES AT A SPEED OF 500 R.P.M. The Upper Diagram Is for the Intake and Compression Strokes; the Lower for the Expansion and Exhaust Strokes

the use of baffle-plates. In most instances pressure-feed crankshafts are provided with an oil-hole in the crankpin radially outward, in which position the extent of the oil discharge is affected greatly by bearing clearances.

Let us refer for a moment to two diagrams in Fig. 15 which indicate the pressure distribution on the crankpin bearing throughout the cycle at a speed of 500 r.p.m. It is evident that, throughout the greater portion of the cycle, the pressure on the pin is radially outward in the region of the inside of the crankpin, thus locating the clearance in the bearing on the outside. As it approaches the center on the compression stroke, and likewise on the expansion stroke, the pressure is reversed.

Fig. 16 gives the pressure distribution at 1500 r.p.m. under full load. The increase in the magnitude of the inertia and centrifugal forces, as the speed increases, is such that the resultant pressure on the pin is radially outward except for an angular travel of 100 deg. during the expansion stroke. It is evident from these figures that, at all speeds and particularly as the speed increases, the pressure on the pin is concentrated chiefly on the inside, thus locating all of the bearing clearance on the outside of the pin. When running under no load, this condition is emphasized. It is clear that, with the oil-hole placed on the outside of the crankpin, the clearance of the bearing will be located in the vicinity of the hole most of the time, thus permitting a free opening through which the oil can be discharged. The oil discharged from the crankpin has a direction tangentially forward from the point of discharge; so, it will be evident that when approaching the upper center the oil discharged from the crankpin is directly in line with the cylinder. Bearing wear and increased clearance will therefore have a marked effect on the quantity of oil thrown from the bearing and the figures that follow point very conclusively to the influence bearing clearance has on oil-pumping and oil consumption.

Fig. 17 is a reproduction of similar pressure-diagrams of eight-cylinder V-type engines, for which the same arguments hold true. To control the extent of oil discharged to the crankpin, it is clear that the oil-hole should be located to maintain a continuous seal regardless of the bearing clearance and the speed; therefore, it should be in the zone of continued pressure, preferably radially inward on the inside of the crankpin.

To appreciate the influence of oil-hole location on the

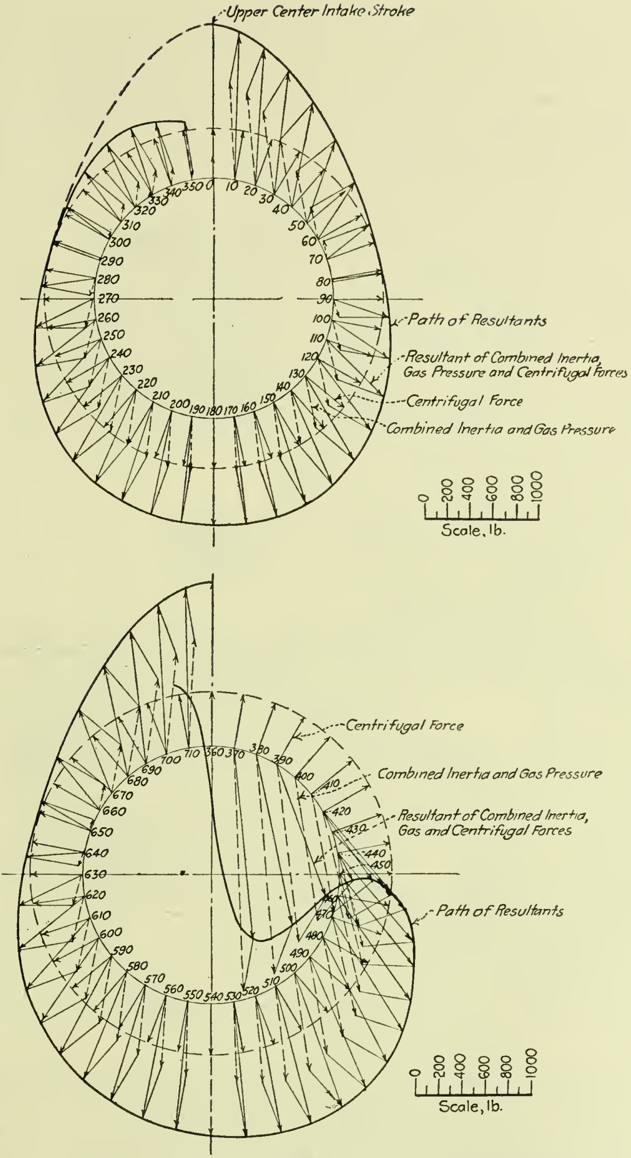
Upper Center Intake Stroke
0 10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200 210 220 230 240 250 260 270 280 290 300 310 320 330 340 350
Path of Resultants
Resultant of Combined Inertia, Gas Pressure and Centrifugal Forces
Centrifugal Force
Combined Inertia and Gas Pressure
0 200 400 600 800 1000
Scale, lb.
360 370 380 390 400 410 420 430 440 450 460 470 480 490 500 510 520 530 540 550 560 570 580 590 600 610 620 630 640 650 660 670 680 690 700 710
Centrifugal Force
Combined Inertia and Gas Pressure
Resultant of Combined Inertia, Gas and Centrifugal Forces
Path of Resultants
0 200 400 600 800 1000
Scale, lb.

FIG. 16—CRANKPIN BEARING PRESSURES AT A SPEED OF 1500 R.P.M. The Upper Diagram Is for the Intake and Compression Strokes; the Lower for the Expansion and Exhaust Strokes

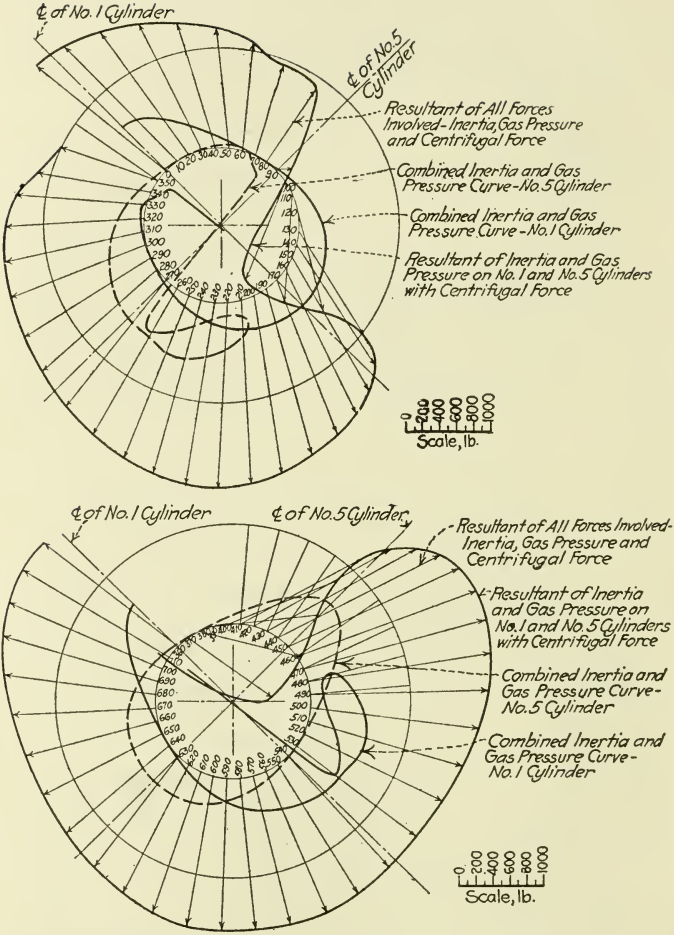

FIG. 17—CRANKPIN BEARING PRESSURES IN AN EIGHT-CYLINDER 90-DEG. ENGINE AT 2000 R.P.M.

The Upper Diagram Is for the Intake and Compression Strokes; the Lower for the Expansion and Exhaust Strokes extent of oil-pumping and oil consumption, compare cylinders Nos. 3 and 4 in the preceding figures, which have standard ring equipment. In all the preceding tests the bearing clearance was increased purposely to approximate service conditions. The comparative quantity of oil as measured from the exhaust discharge or as clearly pictured in the preceding diagrams between cylinders Nos. 2 and 4 and Nos. 1 and 3 is very marked. Fig. 9 shows this condition particularly well. With the same piston-ring equipment in cylinders Nos. 1 and 2, cylinder No. 1 is considerably more affected by a change in the rings. The influence of speed can be observed also. Piston No. 1 becomes noticeably worse, while piston No. 2 is not affected appreciably.

The tests reported in Table 5 are submitted in further support of the controlling influence of crankpin discharge, and the effects of special piston-ring equipment can be observed also. In series No. 3 the oil consumption obtained at 40 m.p.h. with standard ring equipment and the outside oil-hole on the crankpin is given, and also the oil consumption in miles per gallon obtained with the oil-hole on the inside, all other conditions remaining the same. Series No. 3 should be compared with Series No. 1, which shows differences in ring equipment. The use of special ring equipment effected an improvement in some cases but, upon succeeding tests, the oil consumption would increase gradually.

It is interesting to note that increasing the load has the effect of establishing the bearing clearance in the vicinity of the oil-hole when located on the inside through a part of the expansion stroke and, consequently, it exercises a very favorable control on the amount of oil supplied in proportion to the load. From the standpoint of effective lubrication of the pin some arguments can be advanced but, from a consideration of Fig. 18, it would appear that better oil-film distribution is obtained with the inside location of the oil-hole, the influence of centrifugal force tending to distribute the film around the bearing, while with the outside oil-hole the oil is immediately thrown from the bearing. Actual experience has proved that bearing failures at high speed are less pronounced with the inner location of the hole, and that dirt will not score the bearing so readily. Efforts to time the oil supply to the crankshaft through the main bearings properly, so that the discharge from the pin would not occur when the shaft was passing the cylinder,

TABLE 5—OIL CONSUMPTION

Car No.	Type of Rings	Oil-Hole Location	Total Miles Run	Speed, m.p.h.	Oil Consumption, Miles per Gal.		
						Distributing Hole	
					Total	Outside	Inside
Series No. 1							
Experimental	Standard, Old	0	50	40	76.00	......	......
Experimental	New	0	50	40	143.00	......	......
Experimental	New, ⅛ In.	0	50	40	64.00	......	......
Experimental	Special	0	50	40	221.34	......	......
Series No. 2							
G	Standard, Old	0	50	40	139.00	......	......
G, Trial No. 1	Special	0	50	40	300.00	......	......
G, Trial No. 2	Special	0	50	40	240.00	......	......
G	Standard, Old	1	50	40	755.00	......	......
Series No. 3[e]							
58,301	....	..	50	40	......	156	382
O. B.	....	..	50	40	......	151	388
60,887	....	..	50	40	......	156	692
65,629	....	..	50	40	......	294	512

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[e] In this series only the changes in the oil distribution were considered, losses from other sources being neglected.

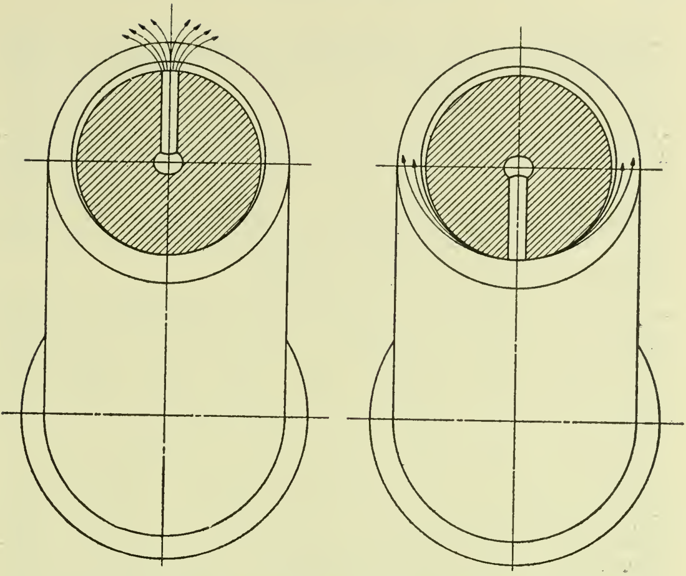

Fig. 18—Oil Discharge from a Crankpin with Inside and Outside Oil-Holes Respectively

proved unsuccessful, because the quantity of oil in the channels of the shaft disturbed the control. The effect of main-bearing discharge should be considered also. It is of advantage to provide a thrower flange adjacent to the rear bearings so that the oil cannot reach the crankpin hole and, consequently, be discharged directly into the cylinder.

An interesting test was conducted to determine the relation between pressure feed and splash feed. An engine was converted to use a splash-feed system, retaining the same piston equipment. It was observed that almost any result could be obtained with the splash feed and, if adequate lubrication were provided for the pin, an excessive quantity was thrown into the cylinders unless a baffle-plate was used. The troughs into which the rods dipped were made so that they could be raised and lowered as desired.

Influence of Oil Viscosity and Effects of Dilution

A reduction in oil viscosity, as a result of an increase in oil temperature, dilution or the use of light oils, will increase proportionately the oil consumption by reason of the additional discharge from the crankpin under the same pressure, unless the oil-hole is located in the zone of pressure so that it exercises a definite control. With an engine so equipped, practically no difference is obtained with the different viscosities produced by the control of the temperature when using the same oil, although a slight addition in the consumption is shown with light-bodied oils, which indicates the effect of cylinder-wall temperature and, consequently, the viscosity of the oil on the cylinder walls and the difference in the piston-ring efficiency with the lowered viscosity.

Fig. 19 indicates the relative oil consumption in proportion to viscosities with different oils, *A* being an oil of 500-sec. viscosity at 104 deg. fahr. and *B* an oil of 650-sec. viscosity at 104 deg. fahr. The third curve is a portion of one prepared by C. W. Stratford,[3] giving the relation between viscosity and oil consumption.

Dilution of the oil by the admixture of liquid fuel is a problem associated with oil consumption. As previously stated, dilution may occur under ordinary operating conditions to the extent that, quantitatively, the fluid in the crankcase is not decreased and the operator assumes that very little oil is being used. The rapid lowering of the viscosity is the dominating influence of dilution which, of course, affects the lubricating value of the oil very considerably and makes it necessary to change the oil frequently to obtain efficient lubrication, unless some one of the types of crankcase-oil refiner that are now being developed is used. The extent of this dilution depends upon many conditions which can, it is believed, be controlled to a large extent.

In the observation of oil consumption and dilution on many engines operating in different parts of the Country, no alarming figures have been obtained, and in very few instances are they in excess of 10 per cent. In the determination of the percentage of dilution, the total quantity of oil or fluid in the reservoir should be known so that, from the percentage of the diluent contained in the sample tested, the total quantity can be computed.

External Oil Leaks and Breather Discharge

External oil leaks are in many cases responsible for excessive oil consumption and often occur at the higher speeds, leaving no direct evidence of the quantity actually lost in this manner. Fog discharged from the breather also has an appreciable influence. This is emphasized

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[3] See Transactions, vol. 10, part 2, p. 86.

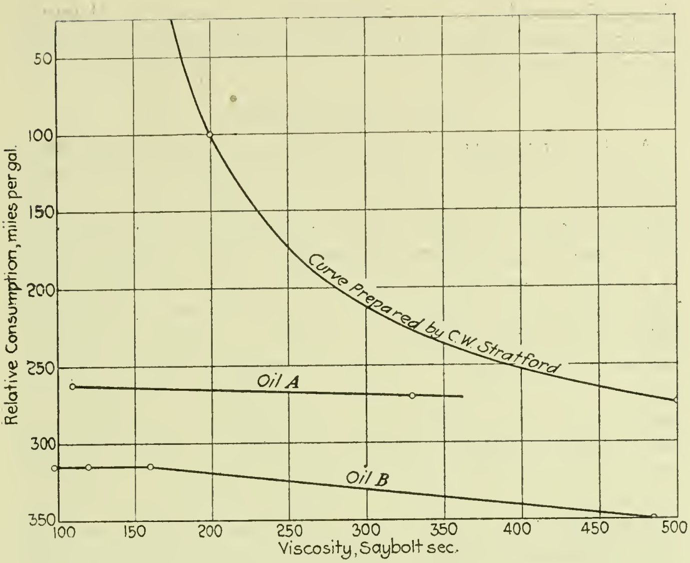

FIG. 19—DIAGRAM SHOWING THE OIL CONSUMPTION WITH RELATION TO THE VISCOSITY AND THE CHARACTER OF THE OIL

particularly where excessive crankpin discharge occurs and also in splash-feed-lubrication engines where light oils are used. Under the latter condition the oil becomes fogged quickly and, if accompanied by piston blow-by, oil vapor will be emitted from the breather.

Controlling the oil pressure in proportion to the throttle opening to effect proportionate lubrication and obtain satisfactory oil economy does not seem to exercise a definite control unless proper consideration is given also to the location of the oil-hole in the crankpin. It is clear that, at the higher speeds, with the increase in the throttle opening and consequently higher pressures, the crankpin discharge will be increased proportionately. It is feasible to use much higher oil pressures with proper oil-hole control which in itself constitutes a favorable argument.

The data presented in this paper cover briefly the general aspect of oil consumption and are supported by actual observations under service conditions. It is believed that the pistons and the rings, when embodying the necessary fundamentals will, in themselves, exercise control over the oil-pumping at the lower speeds and to a definite extent. To insure oil economy under all conditions of operation, it is essential to control the oil supply at its source with a construction that will maintain constant effectiveness.

THE DISCUSSION

A. L. CLAYDEN:—I have devoted much thought to scraper-ring action, because there seems to be so much difference of opinion about it. It appears to me to be possible that in some cases it would work effectively and in others be actually detrimental. The ring mentioned is arranged at the bottom of the skirt so that it runs half out of the bore. It is useless unless it does run half-out, because there is no space for the oil it drives before it to escape. I wish to emphasize that on the down-stroke of the piston the scraping action of the lower one of the several compression rings builds up a very high pressure in the oil-film that is being scraped before it; so, even if the cylinder film is not any too complete, that scraping is enough to insure that the space behind the ring will be completely filled with oil. The depth of the space behind the ring is probably not a very important function, with the amount that is pumped, because if the piston-ring is regarded as a pump piston or as a valve, it will only pass the quadrant, or 90-deg. position, of its stroke. But one can be sure it will be fed full, all it can take; hence the extreme value of the relief holes or grooves immediately beneath the ring and the great value of the ring with sidewise expansion; by "sidewise" I mean fitting the groove expansion.

The crank drilling that Mr. Bull mentioned is probably one of the most important things to study. To some slight extent, a study of the development of various aviation engines was made. It was found to have a very profound effect upon the oil consumption and on the lubrication of all parts of the engine. In fact, all kinds of changes could be made by simply moving the hole around the pin. In many of certain new engines, we finally hit upon a 90-deg. position as giving the desired results; but, of course, a very considerable quantity of oil must circulate for the purpose of cooling.

O. C. FUNDERBURK:—I was connected with the designing of some marine engines ranging in cylinder size from 6¼ to 7¾ in. and developing from 300 to 450 hp. In the multiple-bearing crankshaft we used, it was necessary on account of the high powers to carry oil pressures in each individual crank control. That is, we would have to cut-off the communication all of the way through the shaft, as is apparent in the Liberty engines because of this long crank-arm. We ran this 7¼ by 9-in. engine at 1650 r.p.m. The centrifugal force and the load constitute very large factors in the distribution of the oil, and cause over-oiling in the particular cylinder that has the loosest journal and crankpin bearing. We found it necessary to put a plug in the main-bearing journal of the crankshaft and to make individual crankpins for each cylinder. That greatly decreased the over-oiling in any cylinder where there were loose bearings in proportion to the adjacent cylinder. We found also, as Mr. Bull did, the necessity for moving the oil-hole from the outer position on the crankpin to an inward position. The position we use is 10 deg. from the underneath position. That, we found, made a great difference in the distribution of oil. We also found a great change in consumption due to pressure. We have experimented with pressures of from 10 to 250 lb. per sq. in. We found that a pressure of about 30 lb. per sq. in. gives the highest horsepower with the best oil-consumption. The excess power required to drive the pumps at the exceedingly high pressure and the cooling of the oil absorb sufficient power to show a difference on the dynamometer. We obtained the best oil-consumption in our engines when the oil had a temperature of about 108 deg. fahr. The temperature sometimes ranged from 100 to 120 deg. fahr. with Mobiloil B, and Veedol extra-heavy. The temperature of the oil ran as high as 150 deg. fahr. and there was a marked increase in the consumption. If the oil was colder than 80 deg. fahr., we found an excessive amount of oil on the spark-plugs. This proved very conclusively Mr. Bull's statement that if three rings on the top and one scraper ring are used, the film of oil on the piston could not be discharged on the ring, and the unit pressure between those upper and lower rings would discharge it on the piston. As soon as we got temperatures of the oil as low as 50 deg. fahr. the fouling of the spark-plugs with over-oiling became very much more apparent.

T. J. LITLE, JR.:—I am not a believer in the velvet, or rough, finish on the cylinder bore that some companies have practised, and the rough rings. At least one engine company in this Country has practised finishing the bore, grinding it four times and honing the surface. The final honing operation is done with a stone of very fine grade, leaving the cylinder wall in a polished condition. Many of us have looked into engines after they have been driven several thousand miles and greatly admired the boring. That is the way it should be done at first, and the pistons and rings should not be used as laps to do it.

A thick piston-head is necessary if a cast-iron piston is to operate very satisfactorily in a passenger car. If a thin-headed piston is not used, the heavy piston causes the engine to vibrate excessively. Therefore, I do not believe in using cast-iron pistons.

There has been a very great development in aluminum pistons recently in this Country. I refer to the aluminum-alloy pistons containing about 10 per cent copper and heat-treated to increase the hardness from 75 to 80 up to 175, almost as hard as cast iron. That has an indirect bearing on this whole problem, because when a light piston of great hardness is produced, the ring grooves will not wear. It is when the ring grooves wear that the engine starts to pump oil excessively.

We all know that, if the depth of the ring is increased, its life is increased, for it will not wear the groove wide so quickly. Many companies are careless in fitting the ring in the groove. Some rings are tight and some are loose at the start, right from the factory. The inertia of the ring hammers the groove wide, particularly when the former does not fit tight.

The construction of the ring itself affects oil consumption most. A plain one-piece ring that has a real scraping-edge on the bottom will control the oil-flow. I do not mean the conventional 90-deg. corner, but that if the ring is cut in at an angle and a scraping edge is established at the lower side of the ring, the oil consumption will be changed wonderfully. On a given engine that is consuming oil at the rate of 1 gal. per 300 to 400 miles, the oil consumption can be decreased to 1 gal. per 2000 miles by simply modifying or sharpening the scraping edge of the bottom of all of the rings. The scraper ring does the most good when a line is cut under the edge of the piston. In other words, if the piston is right up to size if the liner under the piston-ring, the lower ring, is the same size as the rest of the skirt, it does little good. But if that point is cut under, its efficiency is increased greatly.

If a little gash is cut in the lower part of the ring at rather an acute angle the scraping effect will be very marked. It is just like that of the ring itself, and it does not provide any oil-holes through the cylinder or employ these in every ring. We experimented on the dynamometer about a month to determine this angle. We varied it every 5 deg. and the difference in the scraping effect was remarkable. With an acute angle I think there is a certain flexing of the edge. In other words, it acts like a chisel going down an oil-stone; you can scrape all of the oil ahead of it, and leave the oil-stone dry. But if a plain piece of metal with a square edge is put down there, it will leave a film of oil.

CHAIRMAN GEORGE E. GODDARD:—I think that the great advantage of that angle feature is that it provides an edge which sharpens itself.

MR. LÍTLE:—I believe that the greatest scraping, of course, is on the lower ring below which there is a gash right through to the interior but no small holes. We had difficulty in getting enough oil through the holes. It is an expensive operation to cut a little slot around the skirt and drill a number of little holes in it. A gash cut right through is, I think, the best construction. It is used very largely on the aluminum-alloy piston and is very effective.

MR. FUNDERBURK:—In connection with the discharge of oil from the rings through the holes, we have had the experience Mr. Lítle relates. We started with 12 No. 40 drilled holes below the scraper hole. That was the third ring from the top. We have increased the size of those holes, and finally we have now 22 holes of 3/16-in. diameter, which, of course, is greatly in excess of the area of the annulus represented by the clearance between the cylinder and the piston.

In a design we prepared for the Government we began some experiments in which we ran the shaft on the same bearing pressure as when the engine was on full power. We found it necessary in connection with the oil pressure to use a refrigerating system to cool the oil, so that we could lubricate at very high speed without scoring; the bearing is 5¼ in. in diameter and runs at 1400 r.p.m., which is far above anything in my experience which had been undertaken in our line before.

J. E. WHITE:—We find in many cases that, if some of the so-called heavy oils are used, the engine cannot be cranked at more than 30 to 40 r.p.m. If an oil that has the proper base is used in winter, one can get as high as 50 or 60 r.p.m. I am speaking of the Packard, the Lincoln, the Cadillac and the Lafayette types of engine.

L. M. WOOLSON:—So far as the minimizing of oil consumption goes, we have found that we can get practically anything we want. We can get an engine to run 5000 miles per gal. of oil if we so desire, or we can do even better than that. But the fact of the matter is that an allowance must be made for sufficient oil-consumption so that the average owner will maintain his oil supply. We have had owners proudly boast about running 10,000 miles and not using a drop of oil. To avoid that condition, we have fixed our engines so that they will use some oil. That is the only possible way of getting fresh oil. We must have fresh oil until we get this dilution problem solved. I think that none of us is really working on this dilution problem as seriously as it deserves. There are two general ways of keeping the oil consumption down, but I think we cannot use either of them until we get the dilution problem solved because a job that will run 5000 to 6000 miles per gal. of oil in the course of the use of a car by the average owner, will be ruined in fairly short order, especially in the cold winter months.

To my mind there are two ways in which we can really control the oil consumption best: by baffles and by scraper-ring construction. We used a scraper-ring construction on high-compression engines with very great success. If we go too low with the oil-consumption we find we get into trouble from hot pistons and hot exhaust-valves. In high-compression aviation-engines some oil must be passed into the combustion-chamber to keep the exhaust-valves cool.

The ordinary type of baffle consists of a slotted plate fitting the cylinder bottom, the rod working through the slot. A baffle like that is worse than useless, because the high velocity of air going by the slot causes a great quantity of oil to be carried with it, the result being that instead of decreasing the oil supply it is generally increased. If the baffles are arranged in the form of semi-circular guards that have the crankshaft center as their center and these guards are extended over the ends of the connecting-rod bearing, the oil can be practically prevented from reaching the cylinder. I do not know why that is not a very much better way than trying to get extremely accurate fits between the piston-rings and the cylinder and the ring grooves, which cannot possibly be maintained during many thousands of miles of travel.

We have been taught for many years that the place to feed the oil is on the slack side of the bearing; yet Mr.

Bull tells us that the place to feed it is on the tight side or, in other words, so that the bearing will plug up the hole. I have had some intensely practical experience with just that construction which enables me to make such a positive statement that, from a bearing standpoint, it is positively the worst thing one can do.

We had a 600-hp. aviation engine on a 50-hr. test. At the end of 25 hr. we found the bearings in pretty bad shape. We then went through this same analysis of bearing pressure that has been discussed and found just what Mr. Bull found. Therefore, we decided to put the hole just where he said we must not put it, on the slack side, so that we would surely get plenty of oil there. We ran that engine another 25 hr. with new bearings and the new oil-hole location but everything else the same, and those bearings stood up. Since that time we have always located the oil-feed holes on all our jobs at 45 deg. leading, which represents the zone of least pressure. That gives us an ample flow of oil through the bearing and helps to dissipate the heat. You can trap the dirt just as well by pressing a tube in the oil-hole as long as desired.

F. E. WATTS:—I think that the principal value of Mr. Bull's paper is possibly not in the conclusions he reaches so much as in the method he follows. I believe it is the method that must be followed in working out the oil problem in any engine. He starts with the oil coming up to a point where it can possibly be drained without giving any trouble. Then he follows that oil through all of the various passages and places it can go to get into the cylinder. I believe that is the way we must lay out any engine. Lay it out on paper and theorize upon it. Possibly, make models and study them as much as you study the engine because, if the engine once gets to running at high speed, so many things happen in it that the different things cannot be segregated.

I believe that the ideal oil-consumption at present is about 1 gal. per 1000 miles. Keeping between that figure and 1 gal. per 400 to 500 miles is doing pretty well. Using large quantities of oil keeps the grit out of the cylinders and the bearings, and the engines last enough longer so that it pays more than the oil costs. It is perfectly easy to study the combustion-chamber and locate some point in it where the spark-plug will keep reasonably clean even if there is a quantity of oil coming onto it.

CHAIRMAN GODDARD:—In some of our experiments for reducing oil consumption we got so low on the oil that we began to get spark knocking. We found that the amount of oil had been cut down so that the carbon deposit we get from poor gasoline was not moistened. I think that with filtered gasoline we get better results than we do, even today, with hot-spots and the like. Our experience is that there must be a little bit of oil in the combustion-chamber to keep the top of the piston moist.

H. S. McDewell:—As some of you may know, in the Navy Liberty engine a scraper with holes is provided. The oil consumption was reduced as a result of that. The 1700-r.p.m. consumption was cut from 14 to between 5 and 7 lb. per hr. In starting that work we went through a rather laborious research to determine what shape that scraper ring should have, and whether this feathered ring was sufficiently better to warrant the increased cost of production. Therefore, we devised a hot plate and a block of 1-sq. in. section. One edge was right-angled, another edge was at 16 deg. and the third edge at 30 deg. We provided a load such that the pressure would be 8 lb. per sq. in., which is the pressure on the piston-rings. We then measured the thickness of the oil-film, and scraped this slot across the plate, which was maintained at a temperature that we assumed to be practically that of the oil-film. We found that under the same unit-pressure, while there was a difference in favor of the sharp edge, it was not sufficient to pay for the increased cost of producing such a ring. Consequently, we adopted the small square ring, and it provided ample space into which to scrape the oil and ample drain-holes. The size of the hole we used was 3/16 in. in diameter. We used 14 of them. I think the cause of the failure of holes of very small diameter is the high surface-tension of the oil, so that an oil-film bridges across those holes and offers too much resistance to the flow of oil draining out to the other side of the piston.

Particularly in the aluminum-alloy piston with cast-iron rings the thing that must be guarded against is not so much the initial clearance as the differential clearance; that is, the increase in the clearance that is due to the different ratios of expansion, the coefficients of expansion of the groove and the ring. The narrower the ring is, the less the actual change in the clearance will be. Consequently, very much better results are obtained with narrower rings.

In regard to the location of the scraper ring at the bottom of the skirt, it has always seemed to me that this would be very analogous, in the case of steam-engine practice, to installing some device to prevent lubrication of the crosshead. The piston, or the skirt portion of the piston, is the crosshead in the gas engine and it should be lubricated. The same thing applies to the use of baffles to prevent the throwing of oil up from the crankshaft. The real problem is to prevent the oil from working past the piston-rings, and to provide ample lubrication for the crosshead surface itself.

MR. LÍTLE:—Placing the feather edge in the bottom of the ring requires 14 sec. for the actual operation. I would rather determine its value by actually knowing how long and how well it produces on the job than in a laboratory machine such as Mr. McDewell used for that purpose.

I agree that the piston and the edge of the crosshead should be oiled copiously. I think it is not advisable to place guards around the bearings to prevent the oil from splashing up into the cylinder bore. It should be just "slathered" with oil. There should be plenty of oil for the cylinder and plenty of oil to pass to the skirt of the piston. The rings should be used to control the passage of the oil.

My experience with engines that use 1 gal. of oil every 300 to 400 miles is that they are using too much oil. It requires too frequent visits to the service-station to have the carbon cleaned out.

CHAIRMAN GODDARD:—We were ready to put in baffle-plates but found that in using the aluminum piston with the large slot we kept the oil down enough so that we did not need them. Also, we are driving away more cars than ever before due to the railroad congestion and it would not be well to keep cylinder lubrication down in these new cars that many times are abused by irresponsible drivers.

We have, however, provided a place in the cylinder where baffles can be added after 5000, 10,000 and 15,000 miles, to keep the oil consumption down if this becomes necessary. By using a large volume of oil, the oil lasts longer, and it will stay in better condition. Some speakers have commented on the rings with the scraper groove in them being in the bottom groove of the piston. I assume that they mean the lowest groove above the wrist-pin. Our experience has been that, if they are put there, not enough oil gets above them to lubricate the two upper rings satisfactorily. Our practice in using scraper rings has been to put them in the top groove, but we could not always get enough drain-holes in there.

A. A. BULL:—It is unfortunate that in considering this question of oil consumption we are usually inclined to pick out some particular feature, instead of trying to consider the matter as a whole. Consequently, as has been evident, in many instances, changes or modifications that are effective under some conditions prove absolutely ineffective under others.

The thing that counts is what happens in service. The purpose in locating the oil-discharge hole as I recommend is to get a condition that will exist throughout the life of the engine regardless of the bearing clearance. In other words, there is a certain discharge from the crank when the engine is new, and it is desirable if possible to keep that quantity of discharge the same regardless of the bearing fit. It is inevitable that the bearing will get loose. If you have an excessive bearing-clearance there is absolutely no guarantee that the oil will reach the place where the bearing pressure exists.

So far as providing adequate lubrication is concerned, the time that you get the discharge is when you change the direction of the pressure on the crankpin due to the influence of the pressure in the cylinder; the greater the load, the longer the interval will be and consequently the larger will be the supply of oil. Mr. Woolson said that it has been shown that the bearings of some engines do not stand up well. I maintain that was not because there was insufficient lubrication or that the oil was not efficiently distributed, but rather that the temperature reached under these particular operating conditions required a larger portion of oil to be circulated. That may have been because the clearance was inadequate. We must recognize that high temperatures call for different clearances.

We must have a definite amount of lubrication in the cylinders. I grant that absolutely. But again I say that if the lubrication given the cylinder is a definite amount when the engine is first built, and if we agree that it is sufficient, all we need to do is to maintain it. In placing the oil-hole at the top of the pin there is no question that the discharge from the cylinder increases as the bearing wears. If, in order to control the oil going into the cylinder, baffle-plates are placed over the bearings, the effect of which is to do just what we are trying to do with the oil-hole location. I should like to know how we expect, when the engine is new, to get any oil in there at all. When the job is new, it should have the most oil; and, after the engine has been worn, it needs less. With the ordinary construction and oil-hole location, we take steps to give it more oil when it needs less.

On the question of how much oil we should use, I believe that 1 gal. per 1000 miles is good all of the time.

Regarding the question of ring wear and piston hardness, I am a champion of the aluminum piston. Mr. Lítle believes that we shall eliminate the troubles with an aluminum piston as regards the side clearance of the ring when we make the piston as hard as cast iron. My arguments on this are that while we may do things to the piston and rings that will more or less limit the wear, sooner or later it will occur. What should be done, if possible, is to provide something that will take up wear automatically and maintain the condition that we know should exist.

It is argued that making the ring thinner will reduce the force of inertia that is responsible for the wearing down of the grooves. If that is the predominating cause, why is it that the top ring of the piston and the second ring will invariably wear at a considerably greater rate than the rings below them? I have seen instances where, after both top rings were worn as much as 1/32 in., the third ring was in fair condition and the bottom ring in the same piston and subject to the same inertia forces was practically in the same condition as when originally installed. I think that there are some other factors affecting this wear that we do not appreciate fully.

As to properly finished cylinder bores, I used to argue that it is useless to put on a very fine finish and make a nice round cylinder bore, because I did not know just what the shape of the cylinder would be under operating conditions. Subsequently, in the development of pistons, we found that the cylinder became peculiarly shaped under operating conditions. Afterward, we made an engine with cylindrical sleeves of the same thickness all the way round and machined both inside and out. Then we found that whatever we put into the cylinder to begin with was maintained pretty well under operating conditions. Rings and pistons that would not produce a compression in the ordinary cylinder, no matter how well it was finished, would work perfectly well in the inserted-sleeve type of cylinder because, under the conditions in which the engines were operating, we had a fairly constant relation between the ring and the cylinder.

In an experiment made 4 or 5 years ago to determine oil drainage, we drilled a number of holes in a piston immediately below one ring-groove. We put in the ring with just an ordinary mechanical fit, cleaned the piston on the inside and painted it white, so that we could trace the flow of oil through the holes. We made a partition on the bottom to prevent any oil from being splashed inside. We ran this piston in the cylinder until there was evidence of the oil's passing up to the top surface of the piston. We expected, of course, to find that the oil had been scraped off the cylinder wall by the ring and pushed through these holes, but the oil was not there. It did not push through until we had made the ring fit tight in the groove. When you make a large hole or slot such as is used with the slipper type of piston, it makes it much easier for oil to pass through. It will be more effective than ordinary ⅛ or 1/32-in. holes would be under the same running conditions.

I made a fairly definite statement in my paper to the effect that I do not believe piston clearance itself has anything to do with oil-pumping. I think that the fit of the piston in the cylinder does have an effect on the way the ring functions. In that respect I agree that the location of the piston-pin has considerable to do with that because, if you have 0.006-in. clearance and the piston-pin is located near the top of the piston, the angle of the piston in the cylinder will be much greater than if that same piston were provided with a pin in the middle of its bearing face. There is a too prevalent opinion that this question of clearance is the real cause of oil-pumping. Let us consider for a moment the slipper type of piston. It is exposed completely on two sides. The oil-film would come up ½ in. thick if it could, but it cannot. If it be true that clearance in itself will permit a larger quantity of the oil to cling to the walls, then I would say that the slipper type of piston would be a very poor job from the standpoint of oil-pumping. Yet it has proved to be very good and for no other reason, in my opinion, than that it is much easier with a large clearance actually to displace the oil-film or roll it up off the cylinder bore and push it through the holes.

The character of the lower edge of the ring is important. I made a statement to that effect in my paper, although I still argue that it is impractical to put anything in a piston-ring that is likely to lose its efficiency. If you make, as Mr. Lítle says, a more or less feathered edge, which you could do by putting in a groove, it is probable that it will retain its sharp edge if you make the angle acute enough. But, judging from the condition of the rings that I have seen, it is possible that the wear would gradually reduce the efficiency of the edge to a point where it would not exercise the control that it did when it was first installed.

Regarding carbon deposit, it is a peculiar thing that in some territory I have been visiting recently, despite the fact that a large quantity of oil is pumped and much oil is used, I did not find excessive carbon-deposit. With so much oil on it does the piston not become hot enough to form carbon? Or, is the effect due to some fuel-condition that happens to exist in that particular territory? I think the latter is the case, because in Detroit, with the same amount of oil, the carbon deposit has been excessive.

I was asked whether compression is any better with multiple-piece rings than with plain rings. That is so largely a matter of the circumferential ring-fit and the trueness of the bore, that I cannot say authoritatively that there is any benefit to be obtained with the multiple ring at this time. I do believe, however, that a multiple-piece ring that expands vertically and fills the groove up and down is likely to retain compression after long use better than the plain ring.

I agree with Mr. Crane[4] in reference to the necessity for presenting the ring squarely to the cylinder-wall. However, I hardly see how he can attribute to this feature alone the success of the construction to which he refers, because, with the wear and with the 0.009-in. taper of the cylinder that he found after use and under which conditions he states the rings and pistons function perfectly, it is difficult to understand how, under such conditions, the rings could remain square and parallel to the walls of the cylinder.

Referring to Mr. Winchester's comments[5], it is established almost beyond doubt that end-clearance is of no importance. If it were as important as he states, we should be in a serious position with respect to the use of a plain ring because we must recognize that, with the wear on the ring and the cylinder, the gap will increase very rapidly; and, under such conditions, we certainly should require something different from a regular angle or step-cut joint.

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[4] See p. 212.

[5] See p. 218.

In the analysis of the piston-ring pressure, it is shown clearly that there is an advantage in increasing the pressure between the ring and the cylinder, but this in itself is of little avail if side-clearance in the ring occurs; for, while the ring itself may be more efficient in displacing the oil from the cylinder-wall, it cannot prevent its passing around the ring instead of through the oil-return holes that may be provided. Under such conditions of side-clearance, however, and with holes at the rear of the ring-groove, the passage of the oil can be prevented to a large extent.

It is true that the features of construction, which the majority of replacement piston-rings possess, visibly demonstrated to the buyer, rarely have any characteristics controlling oil-pumping, and the importance of accuracy in manufacture is often neglected. It must be recognized, however, that there are factors in the cylinder itself that may nullify the benefits of extreme accuracy in the ring.

I stated that a plain concentric piston-ring when properly installed is generally as good or better than rings having fancy constructions of joints; but, in the final analysis of the functions of a ring and recognizing the vital necessity for properly fitting sidewise in the groove, I advocate the use of a two-piece ring of a construction such as is illustrated by the third ring from the left in Fig. 12, which has the ability to expand sidewise in order to fill the groove properly. I have used rings of this construction made especially for use with aluminum pistons for 3 years and cannot fail to recognize the fundamental necessity for a construction of this kind.

Regarding the reduction of oil pressure on pressure-feed lubrication systems, there is no doubt that this will help considerably after the bearings have become loose and it is to be recommended, but, as stated, the service-man usually endeavors to maintain the original pressure recommended, for safety.

It is to eliminate such conditions that the method of oil control from the crankpin has been discussed at length. With the proper location of the oil-distributing hole sealed by the bearing throughout the greater part of the cycle, the oil discharged from the crankpin is not influenced appreciably by the bearing clearance. The main point is that the conditions existing when the engine is new should be maintained throughout its life, regardless of the wear that inevitably takes place.

With regard to positive control of oil pressure under varying loads, I think that the connection with the throttle is not proper, as it will not exercise any control over the oil consumption unless it is accompanied by the control of the oil discharge from the crankpin as outlined.

While I did not make any definite statements as to the manner in which I believe crankcase dilution can be controlled, I am satisfied that this can be minimized to an almost negligible degree by the use of proper manifolds, preventing the entrance of large quantities of liquid fuel into the cylinder. The difference between manifolds in this respect can be demonstrated easily by providing suitable dams at the entrance to the cylinder and collecting the liquid fuel that ordinarily flows into the cylinder under low temperature and speed conditions. Further, the use of a choker for the carbureter that prevents sucking raw fuel into the manifold is necessary. The large differences in dilution that occur as between different engines can usually be attributed to such conditions.

OIL-PUMPING[1]

By George A Round[2]

Oil-pumping is defined and its results are mentioned. The influence of various operating conditions is brought out, particular reference being made to passenger-car service. The factors that control the rate of oil consumption are described in detail and some unusual conditions are reported. Various features of piston grooving and piston-ring design are mentioned and the effect of changes illustrated. The relative advantages of the splash and the force-feed systems as affecting the development of oil-pumping troubles are set forth and improvements suggested. A new device for reducing oil-pumping dilution troubles is described and illustrated.

Oil-pumping may be defined as the passing of oil into the combustion-chambers of an engine at a greater rate than it can be burned cleanly by the fuel charge. The results are spark-plug fouling and carbon deposits or in the absence of these difficulties the rate of consumption may be sufficient to cause complaints from the owners on the oil cost. The amount of oil that can be burned without trouble in any engine depends somewhat on the oil character but chiefly on the load factor and the correctness of the mixture. For example, an engine in tractor service will burn cleanly a volume of oil that in passenger-car service would cause excessive carbon deposits quickly. Again it is often the case that the engine that carbonizes in city service will burn clean in touring.

Because of the influence of the load factor it is possible to use in the more severe classes of service a type of lubricant that, while desirable from the standpoint of lubrication value and high economy, would cause undesirable carbon deposits in engines operating under more moderate conditions. In cars having a high acceleration rate the engine load factor under normal conditions has become very small. With the rich mixtures commonly used conditions are most unfavorable for burning cleanly any oil passing the pistons. This type of engine is also prone to knock readily with slight carbon deposits due to the high compressions usually employed. Consequently, it has been necessary to reduce the oil consumption in such engines to a point that is undesirably low from the dilution standpoint.

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[1] Semi-Annual Meeting paper.

[2] M.S.A.E.—Assistant chief of the engineering division of the automotive department. Vacuum Oil Co., New York City.

To assure good lubrication and to offset the effects of dilution it is desirable that the crankcase oil be renewed at reasonably frequent intervals. This can be accomplished in two ways; by (*a*) periodic draining of the entire supply or (*b*) the frequent addition of fresh oil to replace that used. In cars that show a very low oil-consumption rate the first method should apply, but in spite of repeated instructions to the owner to drain the old oil frequently, he finds it such a disagreeable task that it is not done as often as it should be, and contamination of the oil to an excessive degree is the result. It is a fact, however, that in many engines the rate of oil consumption increases to an undesirable extent after a comparatively short period of service, and it is in the control of this that we are chiefly interested.

FACTORS CONTROLLING OIL-PUMPING

The amount of oil passing an engine piston in a given time depends upon the following principal factors: (*a*) the amount of oil thrown to the cylinders, (*b*) number of piston strokes, (*c*) the efficiency of the means for piston drainage, (*d*) the ring fit, (*e*) the oil viscosity and character and (*f*) the vacuum in the cylinder. In his paper[3] presented at the Annual Meeting in January, 1922, Mr. Ricardo stated that he had been unable to find that the vacuum in the cylinder had any effect on oil-pumping. There is, however, some evidence to the contrary.

A few years ago an engineer working out a device to prevent oil-pumping sealed the crankcase of an engine against air leakage and maintained a vacuum in it equal to that in the intake-manifold. The tests conducted showed that the oil consumption could be cut in two when the vacuum was maintained. Smoking when accelerating after prolonged idling was eliminated, showing that the amount of oil passing the pistons was negligible. While that method of reducing consumption was impracticable, it showed that the vacuum in the cylinder was a factor affecting the amount of oil passing the rings, particularly in worn engines, and the idea has been applied successfully in another way that will be described later.

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[3] See TRANSACTIONS, vol. 17, part 1, p. 1.

Experience seems to indicate that it is not possible to establish a definite rule for the variation in oil consumption with changes in the viscosity and the character of the oil used. In general the consumption decreases with an increase in the viscosity but in several tests we have encountered the reverse condition, under much the same conditions of design. In the engines in which this occurred the pistons were of cast iron with no grooves or relief on the piston skirt. The ring next above the piston-pin was fitted somewhat loosely in its groove in the bottom of which was drilled a series of return holes. In one engine using force-feed lubrication and one employing splash the lower-viscosity oils invariably showed the best economy. The consumption figures for the first case, a small six-cylinder engine, were 1.37 lb. in 10 hr. for the lighter oil, and 2.10 lb. for the heavier under similar operating conditions. For the second engine, a small water-cooled single-cylinder lighting unit, the consumption of lighter oil was 0.5 lb. in 10 hr. and the heavier 2.6 lb. The consumption figures in both cases were corrected to allow for dilution. The viscosities at 210 deg. fahr. were 45 sec. Saybolt for the lighter oil, and 58 sec. for the heavier. Apparently the resistance of the lubricant in passing behind the ring to reach the return holes is the controlling factor, as with a different form of oil return the behavior of the oils is entirely normal. The character of the oil as affecting its evaporation rate, particularly at high temperatures as in tractor or heavy-duty truck service, also has a bearing on the consumption. However, the more heat-resisting oils are not always desirable because from that very characteristic they may not burn cleanly under moderate load conditions. In the majority of cases a compromise is desirable and that must be worked out largely on the basis of practical experience.

THE IMPORTANCE OF PISTON-RING FIT

Proper piston-ring fit is one of the most important factors controlling oil consumption. When the rings fit correctly, the matters of clearance and provision for drainage become of minor importance. The chief points in regard to ring fit are the amount and uniformity of the tension of the ring and its clearance in the ring-groove. Experience has indicated that ring end-clearance is of minor importance in controlling oil consumption. If the rings are fitted with a minimum gap when new, the increase due to wear will have a negligible effect as compared with the other changes that take place.

Other factors being equal, the amount of oil consumed can be varied through a considerable range by changes in the ring pressure. This was clearly shown by some experiments on aircraft engines that were using too much oil. After fitting the bearings as closely as possible and cutting the pressures to a minimum, it was found that a 30-per cent reduction in the oil consumption could be effected by reducing the contact area of the rings about one-third as shown at the left of Fig. 1. It is interesting to note that this method was equally as effective as the other methods of reducing the area shown in the two central views of this illustration. This is contrary to the common opinion but it was checked several times in different engines.

The amount of pressure is, however, fairly definitely limited by the rapid wear that takes place when the ring tension becomes sufficient to cause all the oil to be scraped off. The pressure to cause this varies with the character of the lubricant and the nature of the cylinder and ring metals. A reasonable working limit, however, in terms of the pressure required to close the ring is approximately 2 lb. per in. of diameter for a ring of ¼-in. width, and the other sizes are in proportion. Uniformity of tension throughout the ring is of maximum importance and many cases of oil-pumping have been traced to the lack of this, due to either defective rings or their being distorted during assembly.

From recent experience, it is apparent that in some cases at least, not nearly enough attention is being given to ring inspection or to care in assembling them on the pistons. For instance, in one group of engines taken down to determine the cause of oil-pumping, a large number of rings were found bearing in spots only and of the new rings in stock, over 25 per cent proved defective. A little more care given to this detail would be of great benefit.

As to the relative merits of narrow and wide rings in controlling consumption, for equal wall-pressures there seems to be no difference. Because of their lighter weight the narrow rings do not wear the ring-grooves so rapidly, a decided advantage in aluminum pistons and one that gives them preference from that standpoint. In cast-iron pistons, a wide ring gives excellent results once it has worn to a fit.

The fit of the rings in their grooves is of prime importance. Our experience indicates that in engines where the amount of oil thrown to the pistons is constant throughout the engine life, as with splash systems, the oil consumption increases directly with the increase in the ring-groove clearance. Therefore, unless the lubricating system and the pistons are designed so that a minimum quantity of oil reaches the piston-rings throughout the normal life of the engine, oil-pumping will increase as the rings wear loose in their grooves.

With cast-iron pistons and rings of moderate width, the wear is not particularly rapid, although in motor-truck work it becomes noticeable in less than a year of service as a general rule. With aluminum pistons the wear is much more rapid and unless the oil supply is controlled effectively by other means, excessive consumption results. To avoid this increase in ring-groove clearance, rings have been made that, due to their design, stay tight in the grooves throughout their life. A typical ring of this kind is shown in Fig. 1 at the right. This ring we have found particularly effective in overcoming oil-pumping under a wide range of conditions.

The benefits of such rings are usually slight when compared with new and tightly fitted rings but show up to a marked degree in long service. One example of this was a 3½ x 5-in. four-cylinder engine that was fitted with rather heavy aluminum pistons having approximately 0.007-in. clearance. Using two plain rings above the piston-pin and one scraper ring, the oil consumption was about 1 qt. per 60 miles of a medium-bodied oil. When the second ring was replaced with a ring of the type mentioned, the distance increased to approximately 150 miles per qt. and remained at that point until the pistons were discarded because of excessive slap after some 8000 miles of service. During this time the clearance of the other rings had increased to a marked degree. Numerous other

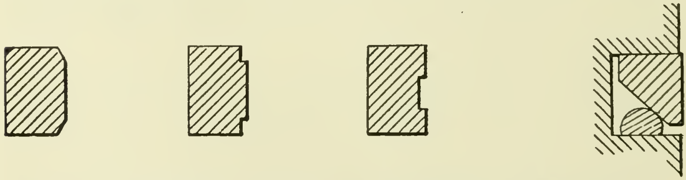

Fig. 1—Examples of Typical Piston-Ring Construction

The First Three Views Beginning at the Left Show How the Pressure of the Piston-Rings Against the Cylinder Wall Is Increased by Reducing the Contact Area, While the Illustration at the Extreme Right Shows a Ring That Is Designed to Remain Tight in the Groove

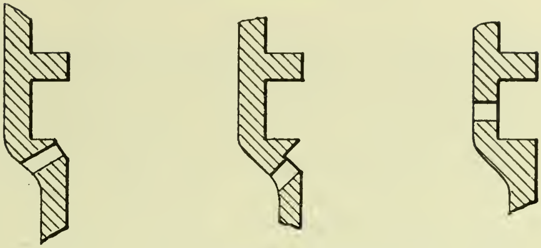

Fig. 2—Some Forms of Oil-Return Grooves

similar cases are on record. While the scraper or skirt ring apparently has only a slight effect in most cases in reducing the oil consumption, it is of value in keeping the wear of the cylinder bore uniform.

Oil Grooving

The use of a properly designed drain groove with adequate return-holes is a great help in keeping oil consumption low, particularly after the rings have become somewhat loose in their grooves. While it has sometimes been the case with new engines that oil-return grooves have not shown much saving, in engines that have become somewhat worn they have proved helpful almost invariably.

Several different forms of return groove have been used. One of the earliest and most common types is shown in Fig. 2 at the left. For this the bottom edge of the lower ring-groove is beveled off slightly and a series of holes drilled from this bevel into the inside of the piston. Where the drain holes are of sufficient size and number and the space for oil collection provided by the bevel is of adequate size, this form of return is effective. It has, however, the decided disadvantage of reducing the area that supports the ring. Consequently, the ring-groove clearance tends to increase more rapidly than it otherwise would, thus defeating the purpose of the groove.

A more desirable form of groove is shown in the center of Fig. 2. With this, a full support of the ring is provided, together with a larger space for the accumulation of the excess oil. In some engines the most effective return has been by holes drilled in the bottom of the lower ring-groove as indicated at the right in Fig. 2. In these the best results have been secured when the ring was slightly loose in its groove and when using a light-bodied oil. Examples of this have already been given.

The size, number and location of the drain holes is an important factor. Holes smaller than 3/32 in. diameter are apparently ineffective unless a large number are used. With 3/32 or ⅛-in. holes, spaced approximately 1½ in. apart, satisfactory results are generally obtained. With certain V-type engines, it has been found that the drain holes are effective only when placed as indicated in Fig. 3. When the holes are drilled on 360 deg. of the groove, the consumption increases. This is an unusual condition and one that has not yet been explained fully but is now under development.

The foregoing comments regarding grooving apply to the more conventional types of cast-iron and aluminum pistons. The latest forms of constant-clearance aluminum pistons are inherently self-draining and should perform satisfactorily provided the rings do not wear loose in their grooves too rapidly and, what is still more important, that the amount of oil thrown to the piston is not excessive.

LUBRICATING SYSTEMS

During the last few years we have seen a gradual trend toward the use of the force-feed lubricating system in which the connecting-rod dip is eliminated. The advent of the V-type engine made the use of this system necessary to assure a uniform distribution to the cylinders. Many have followed this lead in adopting pressure systems for other types, but the results as regards oil-pumping have not always been as happy as desired. The great disadvantage of the force-feed system from an oil-pumping standpoint is that the amount of oil thrown to the cylinders increases in proportion to the wear of the bearings and that, unless special provision is made to offset it, the amount of oil supplied at light loads is far in excess of the requirements. In contrast to this the oil throw with splash systems remains constant throughout the engine life and may be proportioned so that, while the bearings receive ample lubrication, the cylinders are not over-lubricated beyond the ability of the rings to hold consumption to a reasonable figure, even when worn.

Perfection of bearing fits has done much to reduce the oil bleed in force-feed systems, but this is often nullified later by poor repair work and by excessive pressures. In connection with connecting-rod bearing leakage the ideal design uses no shims. Where these are employed either the lead-edged or plain brass shim is satisfactory only when fitted properly. If a bearing fails from lack of oil the lead edge will be destroyed and unless the bearing is refitted by a repairman familiar with the design that fact may be overlooked and trouble may result. For truck and tractor service this shim seems less satisfactory.

In the conventional force-feed system employing a spring-controlled bypass, the capacity of the pump is far in excess of the bearing requirements. Consequently, this excess must be taken care of by the relief-valve, the adjustment of which determines the pressure in the system.

When the car or truck is delivered to the user, the pressure is set to give good results with tight bearings. Once he has the normal pressure, as shown by the gage, fixed in his mind, it is extremely difficult to convince either the user or the average repair-man that the drop in pressure that occurs as the bearing clearances increase, is not a danger signal. It is still more difficult to convince either of them that the proper remedy for the oil-pumping troubles that accompany the pressure drop can be overcome in part by a further reduction of the pressure, without endangering the engine.

In some cases, fixed adjustment relief valves are used and no pressure gages are employed. With such designs wide variations in the actual pressure maintained are

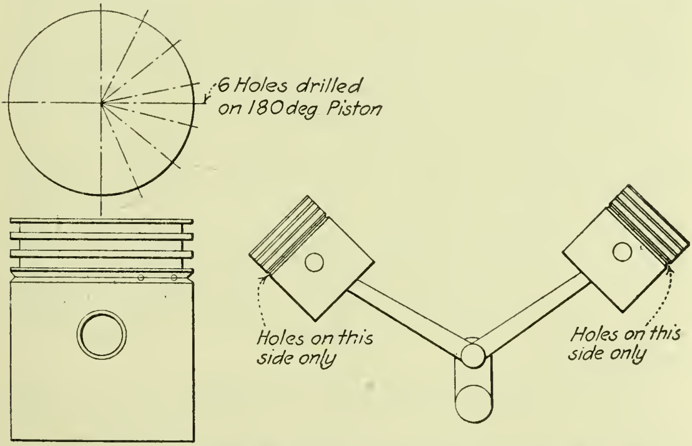

Fig. 3—Drawing Showing the Location of Oil Holes on a V-Type Engine

certain to occur unless the relief valves are carefully calibrated, after assembly. In one case recently investigated, the opening pressure varied from 2 to 25 oz. in 10 valves taken at random from engines and stock. A test showed that the 2-oz. valves provided ample lubrication while with the higher tensions over-oiling was common. More rigid inspection of this important detail would eliminate a source of much trouble and expense to owners and dealers.

The effectiveness of a reduction in the oil pressure in worn engines is very marked, as the following cases show. In the first, an eight-cylinder car, carrying normally about 12-lb. pressure under all conditions of speed and temperature, used about 1 qt. of oil for every 50 miles run and was carbonizing badly. Dropping the pressure to 5 lb. doubled the oil mileage and practically eliminated the carbon trouble.

The other case was a fleet of motor trucks that were giving excessive carbon trouble. The pressure in these engines ranged from 5 lb. when idling to 8 or 9 lb. at governed speed. When the bypass-valves were set to maintain an idling pressure of 1 lb. and a maximum of from 4 to 5 lb., the carbon trouble was eliminated, the average mileage per quart increased from 18.5 to 31.6, and no bearing trouble developed.

The experiences we have had with pressure systems indicate that a far smaller amount of oil is required to lubricate a bearing adequately, and incidentally the cylinders and the pistons, than is generally supposed. Consequently, a marked reduction in the pressure and the volume of the oil supply will be of material advantage in reducing the oil consumption.

It may be contended that to meet the demands of extreme loads and speeds a large volume of oil is needed. There is some question in regard to this, as in many instances we have subjected engines to just such tests with a reduced oil supply, and always without damage. However, if it is felt that this over-supply is needed, some form of throttle or vacuum control of the oil pressure should be used to take care of the prevailing periods of light load.

Another method, which is more simple, is to provide an adjustable bleed in the pump delivery line that will allow the escape of sufficient oil to reduce the oil pressure materially at low speeds, while not preventing the development of high pressures at increased speeds.

A Novel Method of Controlling Oil-Pumping

Earlier in this paper it was mentioned that oil-pumping had been controlled by creating a vacuum in the crankcase to balance that in the intake. This idea has been applied in a very interesting way that not only tends to reduce oil-pumping to a marked degree, but also shows prospects of controlling our dilution and emulsion problems.

Referring to the diagram in Fig. 4, a connection *a* is made to the cylinder wall at a point that coincides with the lower limit of the travel of a groove *b*, located

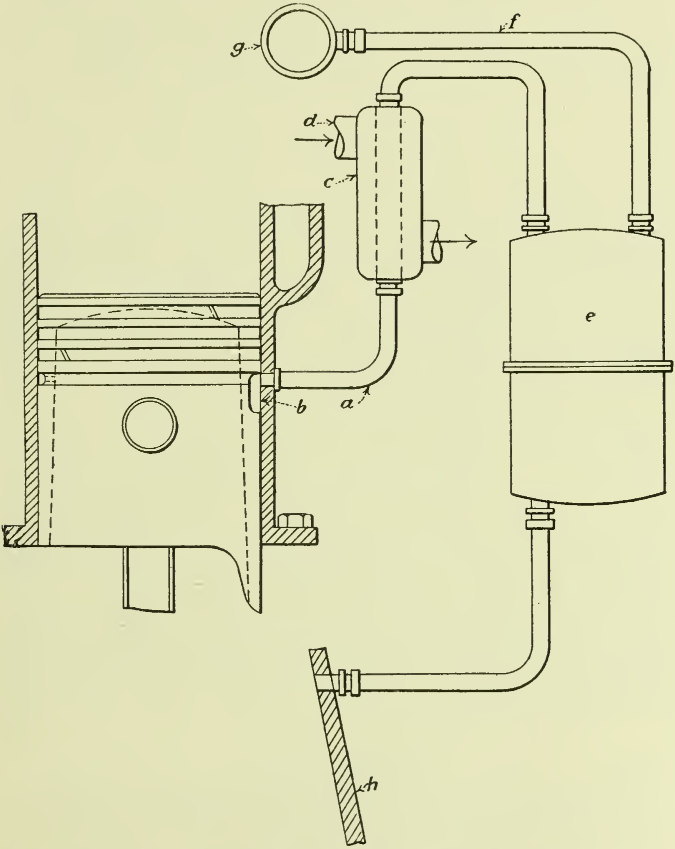

Fig. 4—Diagrammatic Sketch of a Device Developed by the Skinner Automotive Device Co. for Overcoming Oil-Pumping and Dilution

on the piston below the ring next above the piston-pin. A short vertical groove intersecting the circumferential groove and in line with the hole for the connection, is also cut in the piston. The connection *a* leads to a heater *c* that is connected to the exhaust manifold by the pipe *d* and thence to a separator *e*. From the upper part of this device a connection *f* is made to the intake-manifold *g*. The lower part of the separator is cut off from the upper by a valve and leads to the crankcase *h*. On the side of the piston opposite the connection a hole is drilled through the groove into the inside of the piston.

In the operation of the device the suction in the manifold is communicated to the groove on the piston, while the latter is near the bottom of its travel, and during approximately 90 deg. of crank rotation. The vacuum created draws into the heater any oil or oil and fuel mixture that collects in the groove and with it a small amount of air. In passing through the heater the mixture is heated by the exhaust to a temperature of about 375 deg. fahr. On reaching the separator any fuel or water present in the oil is evaporated and carried into the manifold. The remaining oil is passed into the bottom of the separator from which it flows back to the crankcase. To maintain on the other pistons the vacuum that would be destroyed when the upward travel of the piston uncovered the connection to the heater, a lip is cast or fastened on the lower edge of the piston. This keeps the connecting hole covered at all times.

The results of some tests of this device on a number of different cars and trucks of varying lengths of service and conditions, show an increase in mileage per gallon of oil ranging from 100 to 400 per cent. These tests were not short runs, but were carried on over periods of a year or more. As the effect of this device in connection with dilution is to be presented in another paper at a later date, it will only be mentioned in passing. However, the results seem to be very satisfactory and show that we have in prospect another method of solving the dilution problem.

CONCLUSIONS

(1) The lubrication requirements of engines, particularly in passenger cars, do not demand the volume of oil supplied by force-feed systems. Under normal operating conditions the volume of fuel burned is inadequate to consume completely the amount of oil passing the pistons

(2) Oil-return grooving is desirable in all cases and special rings may be required to control the excess oil supply caused by the wear in uncontrolled force-feed systems

(3) More rigid inspection of piston-rings and greater care in fitting them on the pistons will remove a common cause of oil-pumping. Oil-pressure relief valves should be carefully calibrated and set for lower pressures

(4) Some form of throttle or vacuum control is essential with force-feed systems, particularly when used on passenger-car engines

(5) Oil-pumping and dilution can be reduced by a new device that draws from the pistons any excess oil or liquid fuel present, the latter being driven off by heat and delivered to the intake-manifold while the oil is returned to the crankcase

THE DISCUSSION

F. F. KISHLINE:—Were the tests of the device illustrated in Fig. 4 of Mr. Round's paper made on a dynamometer with the engine running near its maximum capacity, or were they made on the road? I have in mind particularly the heat available for cooking the fuel. This might be very low when the engine is running at a low capacity, at which time it has been my experience that the greatest amount of oil-pumping takes place.

G. A. ROUND:—A majority of the tests were made on the road. A number of samples were sent to us from tractors operating in the field on kerosene.

A. L. CLAYDEN:—Has Mr. Round made any tests with the type of piston that is now being produced in large quantities by the Aluminum Manufactures, Inc., in which the ends of the wristpin have considerable clearance and there are two large slots instead of the usual row of holes? Of course, that is not applicable to the suction device; but it is a type of piston that is used extensively now. Has Mr. Round any figures as to how that compares with the more conventional type?

MR. ROUND:—Since the paper was prepared we have received several reports of excessive oil consumption with that type of piston. We believed at the time and still feel that this type of piston will help materially, but where the amount of oil that is going to the cylinders is excessive, we believe that this type of piston will not eliminate oil-pumping. This should be controlled by reducing the oil supply.

H. M. CRANE:—This paper has brought out a point on the question of oil consumption that I do not agree with at all. It states practically that we have to throw very little oil upon the cylinder-walls or take the consequences. I do not think that is the case.

When we first used pressure lubrication, the thought occurred to me that is brought out here, that is, what will happen after the car has been in use for some time and wear and tear has taken place in the bearings? The answer to that seemed to be to flood the bearings and give the oil a free exit. That was done by providing a circumferential groove in the bearing lining, not in the crankshaft, that allowed a free exit from the oil-hole at all times, and by using ⅛-in. shims that did not come anywhere near touching the bearing; in other words, when the engine was stationary in any position, oil could be pumped through the bearing at a rate depending almost entirely upon the oil-hole in the crankpin. It is evident that the amount of oil would be very large, especially as the crankshaft is a very useful oil-pump; in fact, it is the main oil-pump in most pressure-feed systems, due to centrifugal action, the regular pump being used to keep the crankshaft filled with oil.

Different cars that we have built, depending upon the bore of the cylinder, stroke-ratio, crankshafts and bearings, have a pressure on the crankshafts of 2.5 to 25.0 lb. per sq. in. The lubrication is usually adjusted after the completion of the engine by determining the proper pressure to accomplish the required result. Under these conditions cars have been able to run under the same oil pressure in New York City at high speed, in European touring, and in the Alps, with no adjustment for throttling or for varying load. The reason is that the crankshaft, being a centrifugal pump, gives more pressure at high speeds than the direct ratio of power to speed.

With this system the use of return-grooves with holes under the upper rings seemed to be useless, it being difficult to determine any change that was due to the use of such grooves and holes. Sometimes it seemed that more oil went outward through the grooves and holes in the piston than went the other way.

The fundamental thing in the trunk piston is the crosshead. In our experience the length of the crosshead remains approximately the same as the diameter of the piston; with a piston-pin in the center of the crosshead and the walls of the crosshead parallel, we get excellent results. In passenger-car work the use of cast-iron pistons having a clearance of 0.0010 in. per in. of diameter or possibly a little more, gives satisfactory results; for full power at high speeds, up to 0.0015 in. per in. of diameter may be necessary.

Two additions must be made to the crosshead. The first is a set of rings at the top, usually three, as shown in Fig. 5. The top flange is cut away until it will not bear. The second is a ring at the bottom. A land below this ring holds it in place, but it is cut away to such a point that a perfectly free release is left for oil. The cylinder-bore is designed so that in the downward position of the piston the lower ring overruns the bore slightly. This ring pushes the oil down. In a good set of rings the top ring will help to push the liquid fuel up.

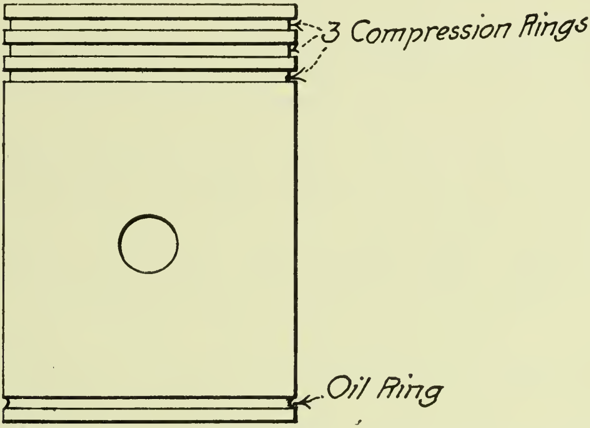

Fig. 5—Piston Equipped with Three Compression Rings and an Oil Ring

Another thing about this type of piston that was wholly unsuspected and accidental is that an oil-film is trapped between the two sets of rings that seems to allow extreme piston clearances, with no piston slap. In aviation practice the pistons appear black after long periods of running, indicating the complete absence of any metal-to-metal contact.

A tapered piston, or one that is too short, does not give a ring a fair chance to do its work. With this type of piston, however, the simplest form of concentric ring of moderate width performs every desirable function, as soon as the ring approaches a bearing on the walls.

Some time ago I installed in a car a set of 4⅜ x 6¼-in. aluminum pistons. These pistons were of the general form already described and had a diametral clearance from 0.008 to 0.001 in. We were able to run with 50-per cent higher oil pressure with this clearance than previously when not using the lowest piston-ring. After running about 40,000 miles, although the performance of the car on the road was entirely satisfactory, the oil consumption very low, the formation of carbon in the cylinders moderate, and the porcelain of the spark-plugs white, we pulled the engine down and measured the cylinders. They were worn to the extent of 0.009 in. in some cases, tapering from the bottom to the top, yet, so far as operation on the road was concerned, they appeared to be satisfactory. The aluminum pistons were replaced by cast-iron pistons of the same design and, barring smoother operation of the engine, it was very difficult to tell the difference. With cast-iron pistons we use an oil pressure of over 20 lb. per sq. in., with a free flow from the crankpin, as has been described, and a crankshaft with three bearings. I cite that simply to show that it is possible to control the oil that is passed by the cylinders if the piston and the ring design is what it ought to be, and that it can be controlled successfully with cylinders that are badly out of shape and with ordinary rings. We do not need trick devices or peculiar joints, or anything of that sort, to get what we want.

Almost half of the piston-ring companies are specializing in special joint construction. If you will figure what the gap in the joints of these rings must have been at the end of 40,000 miles with cylinders 0.009 to 0.010 in. oversize, with normal wear, I think it will be apparent that patented-joint construction is of no use whatsoever and usually ruins a ring that otherwise would wear in a satisfactory manner.

MR. ROUND:—I have had no personal experience with the exact type of piston Mr. Crane describes. It may be that the deciding factor in this design is the relief of the piston below the skirt ring; for, with pistons otherwise similar in design and proportions, we have observed a number of cases of excessive oil consumption. It would seem that the use of this type of piston should be restricted to low-speed engines, because the increased length and weight would be seriously objected to by designers of high-speed engines.

W. L. DEMPSEY:—The proper solution of any problem depends primarily upon an analysis of the cause or causes producing certain results that it is desirable to change or prevent. In the last analysis, there can be but one *efficient* cause for oil traveling from the crankcase into the combustion-chamber of a vertical automobile engine; it must be acted upon by a force in the direction of the combustion-chamber, and this force must be great enough to overcome the friction of the passage of the oil between the cylinder-walls, rings and piston. In addition, it must be great enough to overcome the force of gravity.

It is evident that this energy must be generated either by the engine itself or proceed from some source outside the engine. It is evident, also, that no oil can enter the combustion-chamber during the explosion stroke, because of the excessively high pressures that are exerted in all directions. It is equally evident that no oil can enter the combustion-chamber during the *latter* part of the compression stroke. It is very doubtful if any oil passes into the combustion-chamber during the exhaust stroke, although there may be certain conditions by which a vacuum is created during the exhaust stroke.

We must consider, therefore, only what occurs during the suction stroke of the engine. Let us examine, rather closely, whether there is anything in the construction of the piston, rings and cylinder that would warrant us in believing that oil-pumping is due to energy developed by the engine. First, there is ever present the law of gravity, which tends to carry any oil adhering to the piston or the cylinder back to the crankcase. This tendency requires an actual force to overcome it. If the viscosity of the oil is great, it tends the more to adhere to the spot on which it is thrown by the revolving crankshaft and dippers. This also requires force to move the oil either up or down. Is there any reason to believe that a well-designed piston and ring would carry more oil up against pressure and gravity than it would bring down? I think not. If there is any inherent tendency in the piston and the rings to force oil into the combustion-chamber, why does it not have a like effect upon the unburned fuel that causes so much crankcase dilution?

So long as the piston and the rings are fitted with clearance enough to permit a film of oil between them and the cylinder-walls, there is sufficient space for air and gas pressures to act upon the oil in either direction. Is it possible for the piston-ring to carry up in one stroke out of four more oil than would be cleanly burned at the high temperatures existing during the period of the working stroke? It must be remembered that the force exerted by the revolving crankshaft, the connecting-rods and the dippers is always tangential to the axis of the piston travel, and never parallel to the walls of the cylinder and piston.

Mr. Round enumerates a number of factors controlling oil-pumping, but to my mind the one efficient cause of oil-pumping is the last factor stated in his excellent paper, namely, "vacuum in the cylinder"; or in other words, the force that causes oil to flow upward, against the law of gravity, into the combustion-chamber is the difference between the pressure of the atmosphere outside the chamber and the pressure of the gases within it. The examination of tables, giving the flow of air through an orifice under the influence of vacua of different degrees, may throw considerable light upon the subject.

For the last 3 years the company with which I am affiliated has been equipping cars with an automatic check-valve inserted in the cylinder at the limit of the downward piston-travel. We found that so much vacuum existed at the end of the suction stroke that a valve having a 7/32-in. opening opened with such violence that it was necessary to make the valve and the stem integral and of nickel steel to prevent the head from hammering off; and to deaden the sound caused by the rapid opening and closing of the valves, it was necessary to use a compensating spring or fiber washer. Nor is there any speed at which the valves do not open; hence, we conclude that the vacuum at the end of the suction stroke is more than sufficient to account for oil-pumping.

Mr. Round refers to an engineer who succeeded in reducing oil-pumping by maintaining a vacuum in the crankcase equal to that in the intake-manifold. His paper also illustrates a device, in Fig. 4, that is intended to create a vacuum in a groove around the piston when it approaches its lower dead-center.

Tests that I made on a stationary engine, having a 4-in. bore and a 5-in. stroke, with a light spring and an automatic intake-valve, showed a vacuum, when turned by hand at a rate of 50 r.p.m., of 15 in. of mercury. One-half as much vacuum would be sufficient to pump more than 2 lb. of oil into a six-cylinder engine in a 10-hr. run, if the flow of the oil were not impeded by closely fitting piston-rings. I conclude, therefore, that the principal if not the only factor causing oil-pumping is the existence of low pressure in the cylinder at the end of the suction stroke, supplemented often by a pressure in the crankcase higher than that of the atmosphere.

If we are to cure oil-pumping, let us lessen the vacuum at the end of the suction stroke and at the same time introduce the largest quantity of pure air possible by inserting an automatic air-valve that will unfailingly supply the cylinder with additional air, especially at low throttles, to burn completely and cleanly whatever oil passes into the combustion-chamber.

Mr. Round brought out the fact that they had more trouble on the seaboard with engines pumping oil than in the Rocky Mountain district; that tractors give less trouble than passenger cars; that passenger cars give less trouble when touring than when running in towns and cities. Does not each of these statements bear out the fact that the force that drives the oil into the crankcase is the weight of the external atmosphere? At sea-level the pressure of the atmosphere is 14.7 lb. per sq. in., while at great altitudes it is much less; consequently, there is more pressure or force at sea-level to drive the oil into a vacuum. When a passenger car is touring there is less vacuum, because the throttle is open wider, but when the same car is running in the city, it runs under an almost closed throttle and the vacuum is greater. A tractor when moving without a load requires one-half its engine power for self-propulsion; hence, the throttle is wider open and the vacuum is less than that of a passenger car at low speeds.

The remedy, therefore, seems to be to lessen the vacuum in the cylinder, at the end of the suction stroke, and by ample breathing areas to prevent crankcase compression.

Mr. Round:—I believe that the vacuum in the engine cylinder is in part responsible for the passage of oil into the combustion-chamber, as Mr. Dempsey claims. However, the fact that carbonization, which is a result of oil-pumping, gives less trouble in engines working under fairly heavy loads, is due also to the greater volume of fuel burned, the latter being adequate to consume cleanly much larger quantities of oil than is the case with lightly loaded engines.

J. F. WINCHESTER:—The questions of oil-pumping and oil-consumption are so closely related that the papers by Mr. Round and Mr. Bull[4] can be covered in the same discussion. The experiences of these two men, in some ways, check up with the general experiences that I have had in the last 15 years, 10 of which have been closely associated with various types of oil complaints submitted to one of the large oil companies, but, as a part of the discussion, permit me to present the following:

Mr. Round states that crankcase-oil dilution can be offset in two ways: (*a*) by the periodic draining of the entire supply; and (*b*) by the frequent addition of fresh oil to replace that used. To my mind, the solution of crankcase-oil dilution, from a trouble-preventive standpoint, is the periodic draining of the oil supply, depending upon the number of miles run. This recommendation should be made by local service-men, who understand the type of engine under consideration, for the length of time after which draining will be needed depends upon climatic conditions. In some cases the frequent addition of oil did not overcome or tend to relieve the trouble in cars and trucks that I had under my observation, for the reason that the dilution was in such proportion that it equalled the daily consumption of oil. Therefore, the addition of fresh oil exerted a tendency to flood the engine.

Mr. Round states that ring end-clearance is of minor importance in controlling oil consumption. This statement does not check with my experience. It is my belief that ring end-clearance is a very important factor in controlling oil consumption. In the early days ring-fits were based largely on experience with steam engines in which wide clearances were needed. It is generally conceded that this practice is no longer accepted; that definite recommendations are made by the designers of various types of engine; and that definite clearances are insisted upon in the better-built products. The end-clearance of the ring, to a large degree, depends upon the clearance of the piston and upon the type of work on which the engine is to be used. On the average American engine it will be found that much better results will be obtained from the standpoints of both oil consumption and carbonization if all the rings below the upper ones are fitted with less than 0.003-in. end-clearance. It has been my experience that end-clearance controls the amount of oil that will accumulate under the ring, and has an important bearing on the amount that will be pumped into the combustion-chamber.

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[4] See p. 158.

I have found that the amount of oil consumed is controlled to a large degree by the ring-pressure. In a number of cases where complaints have been made that certain types of lubricating oil were entirely unsatisfactory for a given engine, investigation has proved that the trouble was eliminated by employing piston-rings that had the proper tension and were correctly fitted. Mr. Round states that, in his experience, 2 lb. per in. of diameter for a ring of ½-in. width is the tension that gives the best results. My experience, which apparently checks with that of Mr. Bull, based on Table 3 in his paper[5] which gives ring-pressure values, indicates that a 50-per cent greater pressure, or 3 lb. per in. of diameter of ring, is the proper tension. There have been a number of cases where a ring-pressure of 2 lb. per in. caused trouble, whereas 3 lb. per in. for the respective sizes has been very satisfactory.

Attention has been called to the non-interchangeability of the rings manufactured, and to the indifference in the inspection given many rings before they are placed on the market. This experience checks with mine. I know that very many complaints of crankcase dilution and excessive oil-consumption are brought about by (*a*) the manufacturers missing this important point, and (*b*) the patent-ring manufacturers laying great stress upon the features of construction that they can visibly demonstrate to the buyer, yet neglecting to manufacture the articles within close limits. My experience agrees with that of Mr. Crane who stated that the majority of so-called special piston-rings on the market do not give as good results as the plain concentric ring. Mr. Bull believes that the two-piece ring is the most desirable.

Little stress is laid upon the oil-control groove in the piston-ring test. I have found that with the type of ring shown at the right in Fig. 1 the passing of oil into the combustion-chamber is controlled very satisfactorily. Mr. Bull states that the type of piston-ring having an inherent twist, so that the unit pressure on the bottom edge is in excess of that on top, may be beneficial. It is appreciated that rings of this type can be manufactured, but there is a serious doubt in my mind whether this theory will work out in practice. If a ring of this type has proper clearance in a groove, this inherent twist is controlled. Therefore, it fails to come into action. I have tested a large number of rings of this type and fail to see wherein they give better results than a properly fitted plain ring.

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[5] See p. 174.

Considerable stress is placed upon the operation of the lubrication system. There is no doubt that, if a force-feed system becomes worn, there is a decided tendency for an excessive amount of oil to be thrown into the piston and to find its way into the combustion-chamber in various ways. This difficulty is particularly noticeable in so-called high-speed engines operating in city traffic. Under these conditions, a drop in the original pressure becomes necessary. As brought out by Mr. Round, the average manufacturer or service manager is inclined to insist that the original pressure recommended in the instruction book supplied by the manufacturer be adhered to. Careful instructions covering this point, by various manufacturers to their service-stations, would result in many of the present-day difficulties being overcome. Illustrations of the variations of individual spring-pressure furnished with pressure-feed systems, as brought out by Mr. Round, coincide with those of my experience. It is surprising to find the number of manufacturers of high-grade equipment that overlook this important point by failing to calibrate properly the tension of the springs of the lubrication system.

Mr. Round advocates the positive control of oil pressure under varying loads. I agree with him on this and submit that the best possible control is that which is connected with the throttle.

The question of employing auxiliary devices to control crankcase dilution or oil-pumping is naturally one that requires research work. But this difficulty could be taken care of properly by the further improvement of the oiling system of today, without the addition of auxiliary devices that tend to complicate the car's mechanism.

In the discussion Mr. Round advocated abandoning the use of shims. I believe this would work a hardship on the average American manufacturer. It would also result in increased up-keep cost in car and truck operation. The trouble that Mr. Round cites is brought about through a lack of education on the part of the mechanic. Therefore, education is what is needed, not a change in manufacturing methods or design. Neglect on the part of manufacturers to educate properly a suitable number of all-round mechanics is resulting in our puzzling over problems that we should never meet. The industry as a whole probably is suffering more from this cause than from any other.

My recommendation is that proper steps be taken to provide the future generation with a sufficient number of trained men to maintain economically the types of car being put out at the present time. We should not abandon present-day practice, which has accomplished economical results from the standpoint of both manufacturing and up-keep, because mechanics have not been educated sufficiently to make the proper repairs.

Great improvement in oil-pumping and crankcase dilution would be effected if manufacturers, service-men and instructors dwelt more thoroughly upon the subject, and if the average garage-man were able to buy standard piston-ring gages and tension testers that would enable him to determine the quality of the product before it is applied. He should be supplied also, by the car builders, with detailed instructions regarding the fitting of shims and other parts that have an important bearing upon the results to be obtained after the car has been overhauled.

MR. ROUND:—Mr. Winchester's point regarding the overcoming of crankcase-oil dilution is well taken, but there are cars in operation in which oil is consumed so rapidly that replenishing the oil supply is sufficient to maintain the oil in reasonably good condition, thus offsetting any excessive dilution. This, however, is not the general experience.

Regarding the education of service managers and mechanics, too much emphasis cannot be given to this. Even today, we find branch-office service-managers, to say nothing of foremen and mechanics, who do not believe that the thinning out of crankcase oil is due to fuel dilution, and who show an equally meager knowledge of other fundamentals. A series of service bulletins to reach not only the managers but the individual mechanics as well, written in an interesting way, would be a great help along this line.

OVERHEAD CAMSHAFT PASSENGER-CAR ENGINES[1]

By P M Heldt[2]

The gradual trend toward overhead valves in automobile engines, as indicated by an increase in their use on American cars from 6 per cent in 1914 to 31 per cent in 1922, has been accelerated, in the opinion of the author, by their successful application to aircraft engines and by the publicity given them by their almost universal adoption on racing machines. Tractor engines recently brought out show the advantage of this construction. Methods of operating valves in the cylinder-head; the advantages of the valve-in-head construction as regards the form of combustion space, engine cooling and high-speed operation; the reason for using an overhead camshaft to operate the valves on racing engines, the question of noisy operation and the possibility of having an overhead camshaft engine operate as quietly as one in which the camshaft is enclosed in the crankcase; the location of the drive in the various foreign engines of the overhead camshaft type; the silent operation that is possible with a rear drive; the use of chains and spur, helical, worm and spiral bevel gears for the camshaft drive, with the advantages and disadvantages of each method and descriptions of specific applications; and some radical designs of overhead camshaft drive and valve-actuating mechanism that have been developed abroad are among the topics discussed. The three methods of operating the valves: (*a*) directly through the action of cams on followers secured to the end of the valve-stem, (*b*) through the interposition of single-armed levers or adjusting blades between the cams and the valve-stems, and (*c*) by the use of tappet levers, are also outlined with particular reference to the specific applications of each. Numerous illustrations supplement the text.

For a number of years there has been a gradual trend toward the use of overhead valves in automobile engines. This movement probably was accelerated by the success of aircraft engines with this valve arrangement, as well as by the publicity given to this construction by its almost universal use on racing engines. That valves located directly in the cylinder-head are not merely a fad but possess practical advantages can be concluded from the fact that practically all the tractor engines brought out during recent years have the valves so located.

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[1] Semi-Annual Meeting paper.

[2] M.S.A.E.—Engineering editor, Class Journal Co., New York City.

Valves in the cylinder-head can be operated either by tappet-rods extending up one side of the engine or directly from an overhead camshaft. A third method, which consists in arranging the valves at right angles to the cylinder axis and operating them from a camshaft in the crankcase through the intermediary of long, double-armed levers, is exemplified in the Duesenberg engine. This arrangement necessitates the provision of a valve-pocket on top of the cylinder-head, and in most respects, especially as regards the form of the combustion-chamber, resembles the L-head engine more than the true valve-in-head engine which has no valve-pockets.

The advantage of the valve-in-head engine over the L and T-head types is that the combustion space has a much more favorable form. It has less cooling area in proportion to the volume than either of the other types. Consequently the heat loss to the jacket is reduced, and the distances from the firing points to the most remote part of the firing chamber are smaller, so that the flame is propagated through the combustible charge in a shorter interval of time. This tends toward increased power and higher fuel economy. The decrease in the loss of heat to the water-jacket due to the simple form of the combustion-chamber also facilitates the problem of effective engine cooling. The combustion-chamber, the portion of the cylinder casting subjected to the highest temperatures, is more symmetrical and the cylinders therefore are less likely to be distorted by unequal expansion. There is also less likelihood of the presence of masses of metal in the upper part of the cylinder casting that would be difficult to cool, as well as of the formation of steam pockets.

While considerable emphasis is laid on the advantages of the valve-in-head construction from the standpoint of engine cooling, the majority of designers who have adopted the practice have done so because they expected increased engine power and reduced fuel-consumption. The inlet as well as the exhaust-valve passages can be made shorter and more direct than in any other form of cylinder, and the overhead-valve type of engine lends itself particularly well to high-speed operation. It is also claimed that there is less tendency to knock. In view of the greater ease of effectively cooling the combustion-chamber this claim does not sound altogether unreasonable, but no definite proof of it seems to be available.

The tendency toward the overhead-valve type of engines is reflected by the analyses of car specifications that are being made annually. In 1914 only 6 per cent of the cars listed had such valves, while in 1922 the percentage had increased to 31. This applies to American cars. In Europe the valve-in-head engine has made progress, particularly during the last year. Disregarding engines of the sleeve-valve type, 32 per cent of the European models listed in 1922 had overhead valves, as against 20 per cent in 1921. British companies seem to be in a state of transition, many of them offering cars this year with both valve-in-head and side-pocket-valve engines.

Tappet-Rods and Overhead Camshafts

In most engines of the valve-in-head type used on passenger cars the valves are operated by tappet-rods, whereas in most racing and aircraft engines they are operated by overhead camshafts. It is interesting to analyze the reasons for this difference. The side-rod method of overhead-valve operation is the older, at least in the sense that it was the first to gain much popularity. In one way it is also the simpler and more natural, for all that is wanted is a reciprocating motion, and what means of transmitting a plain reciprocating motion over a moderate distance could be simpler than a push-rod? However, when engines attain high speeds the problems arising from inertia effects must be considered, and in order to permit high-speed operation overhead valves are generally employed. Increased power output results directly from the increased operating speed and the gain in economy is also in part dependent upon it.

In racing and ultra high-speed engines the speed of the engine is limited by the inability of the valves to close promptly at extremely high speeds. The inertia of the reciprocating parts of the valve becomes so great that it is practically impossible to get the cam followers to follow the contour of the cams. The force necessary to close the valves that is furnished by the valve-springs becomes greater in direct proportion to the weight of the reciprocating parts of the valve. In one sense there is almost no limit to the force that can be obtained from a spring, but in practice, if the pressure of the valve-spring is increased, the stress in the material of the spring generally is increased also with a consequently greater risk of breakage of the spring. A very stiff spring puts great stresses upon the valve itself. In order to permit higher operating speeds it is generally a much better plan to reduce the weight of the reciprocating parts of the valve than to increase the stiffness of the springs. To reduce the stress on the valves and their springs at extremely high speeds, motion should be transmitted in the rotating form as close to the valves as possible and then converted into a reciprocating motion. This is the reason that racing engines are so frequently provided with overhead camshafts.

Another aspect of the valve problem that needs consideration, particularly in connection with its application to passenger-cars, is the requirement that engines shall operate without appreciable noise. One cause of noisy operation is the clearance that must be given the valves, particularly the exhaust-valves, when they are cold. The need for this clearance is due to the fact that as the engine heats up the exhaust-valve stem expands much more than the engine as a whole and must be given freedom in which to expand or the valve will not close; a condition that is generally followed by disastrous results. If the cams are directly over the valves the chances of unequal expansion and the consequent need for clearance are greatly reduced. In this connection the relative temperatures of the cylinder-block and the valve-rods at the side are factors. If the cylinders are air-cooled and the valve-rods are located so that they are not greatly influenced by the heat of the cylinders but remain relatively cool, it is conceivable that the cylinders will expand much more than the rods. This difference in expansion has the same effect as the expansion of the valve-stems, and additional clearance must be allowed the valves. On the other hand, if the engine is water-cooled and the valve-rods extend through enclosed spaces within the cylinder-block, the rods and the block will be of substantially the same temperature and there will be no difference in expansion. Noiseless operation is not important in racing and aircraft engines.

There seems to be a general impression that it is difficult to make overhead camshaft gears operate noiselessly. Just why this should be so, if it is so, is not immediately apparent. Noise in cam gearing depends on the one hand on the cam outline, which could be made the same as in engines with the camshaft in the crankcase; and on the other hand on the mass of the reciprocating parts, that, as already pointed out, can be, and usually is, considerably less than in a valve-in-head engine in which the valve-rods extend up the sides. It may be that the impression was created in the early days when overhead camshafts were exposed. In modern engines of this type not only the camshaft but the whole valve mechanism is enclosed, so that it should be as easy to render an engine of this construction noiseless as one with the camshaft in the crankcase.

If there is any justification for the belief, the reason probably is that there are generally more wearing contacts in overhead camshaft gearing than in the more conventional type. At each of the contacts wear will occur and the play or slackness in the camshaft operating train therefore will increase more rapidly than in the ordinary L-head engine. Every time the nose of a cam passes from under the cam follower there is a tendency, which is more pronounced in the case of a four-cylinder engine than in one with more cylinders, for the camshaft to jump ahead and take up the slack of the drive, thus creating noise.

In connection with the older engines with side tappet-rods the rapid wear of the tappet-lever bearings is unpleasantly remembered. The only means of lubrication of these tappet-levers was an oil-hole and the excessive wear was probably due not so much to the inadequacy of the lubricating means as to neglect to make use of them. Presumably with the newer engines of this type having wick or even force-feed lubrication this difficulty has been overcome. With some of the overhead camshaft engines tappet-levers are also used, but they are now always enclosed and therefore can be effectively lubricated. The difference is therefore not one of camshaft location but rather of valve actuation and details of construction. Another thing in favor of the overhead camshaft is that it generally leads to a very symmetrical engine with the accessories so located that they are quite accessible and with the sides of the engine comparatively free from obstructions.

One method of judging an overhead-valve engine is by the facility for removing the cylinder-head for cleaning out the carbon by scraping and for grinding the valves. In an engine with the camshaft in the crankcase, the amount of disassembling and reassembling necessary for these two jobs is usually the same, but in the case of an overhead camshaft engine, this does not hold true. In many designs the whole camshaft and overhead valve-operating gear must be taken apart before the valves can be ground in, and this, of course, is a rather serious objection.

From the standpoint of manufacturing cost the overhead camshaft is not a very attractive proposition. The provisions that must be made for the complete, oil-tight enclosure of the camshaft for lubrication, the enclosure of the drive between members, of which one is carried by the crankcase and another by the cylinder-head, and the lubrication of this drive, involve considerable expense. There are ways of producing overhead camshaft engines comparatively cheaply but these engines are likely to prove unsatisfactory.

The overhead camshaft engine has not yet made much headway in this Country, only four cars that are at all well known having such engines at present. In Europe, on the other hand, many of the leading makes now employ this type of engine. These include Lanchester, Napier, Wolseley, Straker-Squire and Leyland in England; Bugatti, Farman and Hispano in France; Ansaldo and Diatto in Italy and Mercedes and Szawe in Germany. As a passenger-car engine the overhead camshaft type is practically an after-war product, but motor-truck engines thus equipped were used by several companies in Germany prior to the war. I assume that the four American overhead camshaft engines, the Wills-Sainte Claire, the Leach Biltwell, the Duesenberg and the Frontenac, are well known, as all have been described in the technical journals within the last several months; consequently, only overhead camshaft engines of foreign design will be considered.

LOCATION OF DRIVE

In connection with the drive the question of its location at the front or rear comes up. It is the almost universal practice to have the camshaft drive at the front and this location is practically standardized. It is well known that trouble has been experienced with camshaft drives in the last few years and there is suspicion that the fact that the drive is at the front, where it is subjected to the effects of torsional vibration of the crankshaft, has something to do with it. Since the introduction of engines with six and more cylinders running at high speeds a great deal of trouble has been caused by torsional vibration. Owing to the periodic impulses in a gasoline engine, the engine naturally tends to accelerate and decelerate periodically and the flywheel is provided to minimize these speed fluctuations. The greatest fluctuations of speed occur at the forward end of the crankshaft. If the natural period of vibration of the shaft happens to coincide with the periodicity of the impulses received by this end of the shaft, a periodic torsional rocking motion is superimposed upon the regular rotary motion of the shaft. After each explosion in one of the forward cylinders the crankshaft will jump ahead, only to snap back the next moment. This effect, of course, occurs only at certain critical speeds of the engine and in engines with exceedingly stiff crankshafts these speeds may never be reached in regular operation. Where a critical speed is reached in the regular operation of the engine it is obvious that such a condition must be rather hard on the camshaft gears if they are located at the forward end.

Mercedes for many years has located the camshaft gears at the rear as shown in Fig. 1 and another German maker, Szawe, has recently followed this example. If there is any part of the crankshaft that rotates more uniformly than the rest, it is the part at the flywheel. This, therefore, would seem to be the best part from which to take the camshaft drive. The objection has been made that this location renders the camshaft gears much less accessible. This is true, but if it renders the gears at the same time invulnerable, what does it matter? With the light work the camshaft gears really have to do it should not be difficult to provide a set of gears that would last the life of the engine. In that case the question of accessibility would not arise.

I believe that anyone contemplating the production of an engine with an overhead camshaft should give the location of the camshaft driveshaft the fullest consideration, especially if the engine has six or more cylinders. From the standpoint of silent operation also the rear drive has something to commend it and conditions are still further improved when the fan is driven from the forward end of the camshaft, making the resulting or total torque on the camshaft non-reversible and thus eliminating the cause of noisy operation referred to in the foregoing.

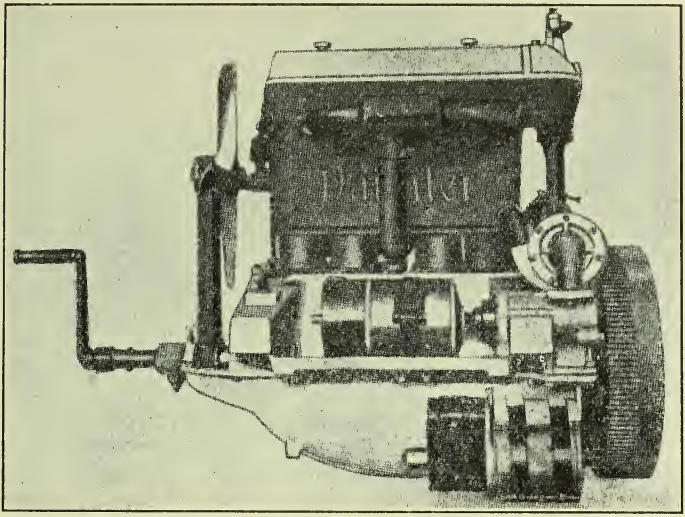

FIG. 1—MERCEDES OVERHEAD CAMSHAFT ENGINE WITH REAR END CAMSHAFT DRIVE

SILENT CHAIN AND SPUR GEAR DRIVES

One method of driving an overhead camshaft is by silent chains and this has been used in a few cases for many years. One of the four American overhead camshaft engines, the Frontenac, has this form of drive. From the manufacturing standpoint the chain drive is very attractive, as apparently all that is required is a sprocket wheel on the crankshaft and another on the camshaft and a chain to connect the two. In practice however various difficulties crop up which have militated against the more extensive use of chains. A single chain between the crankshaft and camshaft is unsatisfactory because it would have to be abnormally long, and with the wearing of the links or the "stretching" of the chain it would whip and be likely to hit the wall of the casing, unless the latter was made with a large clearance space. Two chains therefore are generally used, one transmitting the motion to a short intermediate shaft, which may be the fan shaft, and the other transmitting it from this shaft to the camshaft. This does not dispose of the question of chain adjustment however. One way of solving this problem consists in driving not directly to the camshaft but to a short shaft at the end of it that drives the camshaft through a Hookham joint, and then providing both this short shaft and the intermediate or fan shaft with an eccentric adjustment. Another method consists of the use of chain-tightening idlers.

In the British A.C. six-cylinder engine the camshaft drive is by a long silent chain direct from the crankshaft to the camshaft, with a spring-loaded jockey pulley to take up the slack. The drive is at the rear end of the engine, as is shown in Fig. 2. The ill-effects of the intermittent torque that the chain must withstand, even in this six-cylinder engine, are reduced in a degree by reason of the water-circulating pump being located at the end of, and driven directly by, the camshaft; the load represented by the pump being positive and uniform puts the torque always above the zero line, and though it does not eliminate or even reduce the torque variations due to cam-succession it maintains a positive pressure between the teeth of the sprockets and the chain. For this reason such a pump location seems to be worth considering, particularly on four-cylinder engines, no matter what form of drive is used.

On the Wolseley engine there is a short silent chain for transmitting power to an intermediate shaft directly above the crankshaft. Because of the shortness of this chain the troubles of whipping, and the like, are absent and no means for tightening the chain are required.

The spur-gear drive has given excellent service on racing engines. This involves a train of spur-gears extending up the front of the engine, and one would expect it to be rather difficult to make such a train run quietly, even if helical teeth were used. In racing engines the shafts of the intermediate gear are mounted on ball bearings, and this would no doubt also be required in high-class passenger-car engines, which would make the drive rather expensive. Another objection to the spur-gear train is that, since there are a number of wearing surfaces, when the gears begin to show serious wear a considerable backlash will soon develop. It would then probably be almost impossible to keep the drive quiet. The gears might be so liberally proportioned that the wear would be a negligible quantity, but the high cost and the rather unsightly appearance of the large gearcase would remain. An advantage shared by the two methods of drive is that they require no thrust bearings, whereas all the other more common forms of drive require such bearings. These involve considerable expense, including not

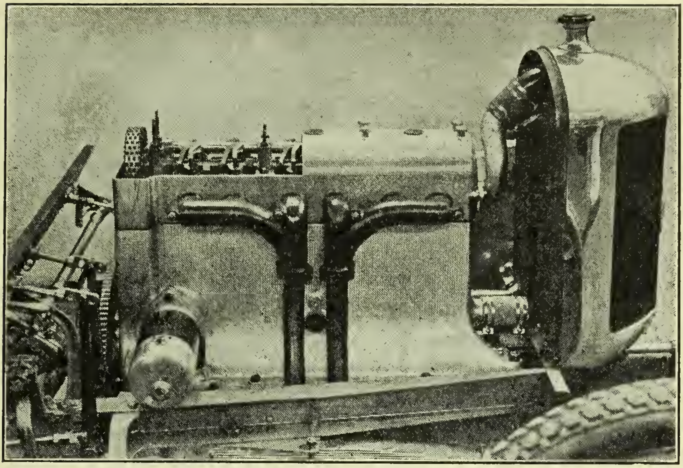

FIG. 2—BRITISH A. C. ENGINE WITH CHAIN-DRIVEN OVERHEAD CAMSHAFT

only the cost of the bearings themselves but also that of machining the mountings and fitting the bearings.

A form of drive for overhead camshafts that has met with success and is now used on several well-known makes of car is that through helical gears with axes at right angles, or the worm drive. Formerly the so-called spiral gears were used to a great extent for the camshaft drives of stationary gas engines. Their most valuable feature is their silent operation. If made of liberal size and provided with adequate means of lubrication they give satisfactory service, but as there is theoretically only point contact the unit pressure at the contact surface is quite high and if the lubrication is not dependable the backlash is likely to increase rapidly. For this reason several British manufacturers use true worm gears for the drive. Worm gears differ from helical gears in that the face of the wormwheel is throated, conforming to the circumference of the worm, whereby an increased bearing surface is obtained. One advantage of the helical gear or worm drive is that the vertical shaft is offset from both the crankshaft and the camshaft and therefore can have extensions for accessories coupled to it at both the top and the bottom. With a bevel-gear-driven intermediary shaft extra gearing is required for accessories at least at the bottom.

It is probably between the worm gear and the spiral bevel gear that the chief competition for supremacy in the overhead camshaft field will arise. The strong and the weak points of the two are practically the same. Both are absolutely quiet in operation, which constitutes their strong point, but both require means for the accurate adjustment of the mesh if they are to work satisfactorily. The advantage of worm gear in respect to greater handiness for the driving of accessories in certain locations is offset by its somewhat lower efficiency and lack of symmetry.

An overhead camshaft is one of the features of the six-cylinder aluminum engine brought out by Napier of England some years ago. The camshaft is hollow and is supported in seven cast brackets bolted to the aluminum cylinder-head as is shown in Fig. 3. It is located centrally above the cylinders. The valves are staggered, one being on one side of the camshaft and the other on the opposite side. There are two rocker-shafts supported on the same brackets as the camshaft. The rockers or valve-levers extend underneath the camshaft and are arranged so as to give a lift to the valves greater than the height of the cam nose. The camshaft drive on this engine is by worm gearing. The vertical shaft is made in three parts: one carried by the housing at the forward end of the crankcase; the second carried by the cylinder-head; and the third an intermediate shaft secured to the top and bottom parts by flanged couplings. From the bottom of the vertical shaft the oil-pump is driven, while the generator and magneto drive is taken off at the joint between the bottom and the intermediate section, the worm being combined with a coupling flange.

In connection with the Napier camshaft drive a description may be given of its camshaft brake, a feature found only on a few high-grade engines. A camshaft brake may be more necessary in an engine having a worm drive for the camshaft than in an ordinary L-head engine with direct spur-gear drive to the camshaft, owing to the fact that the worm drive might permit more backlash in the drive. The object of the camshaft brake is, of course, to prevent the camshaft from snapping ahead as the nose on any particular cam passes its follower. The brake consists of a metal disc mounted in the housing of the worm gear in such a manner that it cannot rotate but is forced against a flange on the rim of the worm gear by a coiled spring in a mounting hub bolted to the worm-gear housing from the outside. Adjustment

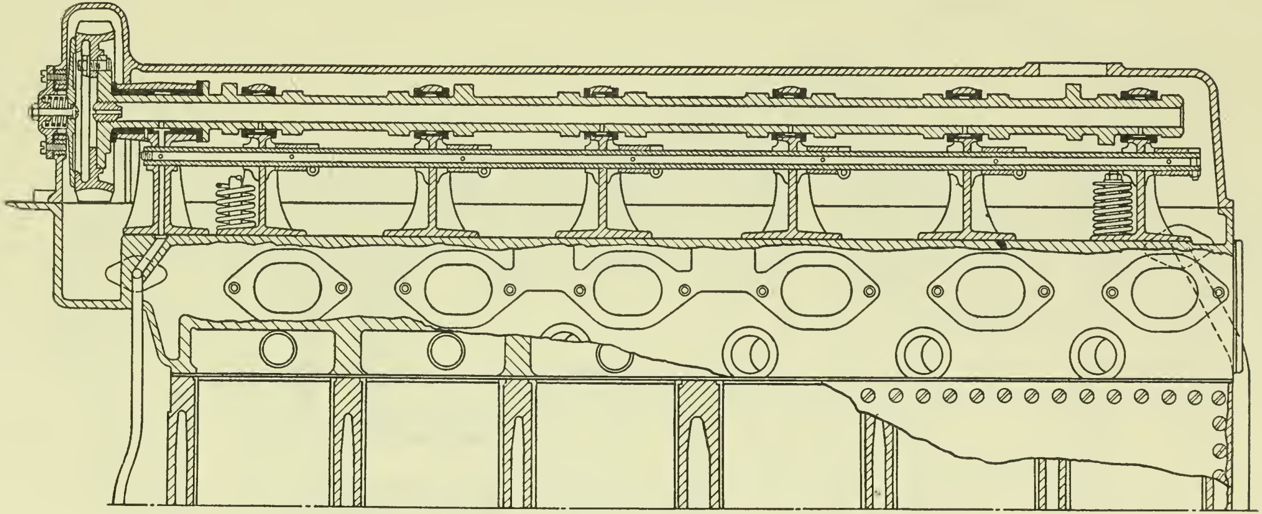

FIG. 3—THE UPPER PORTION OF THE NAPIER OVERHEAD CAMSHAFT ENGINE

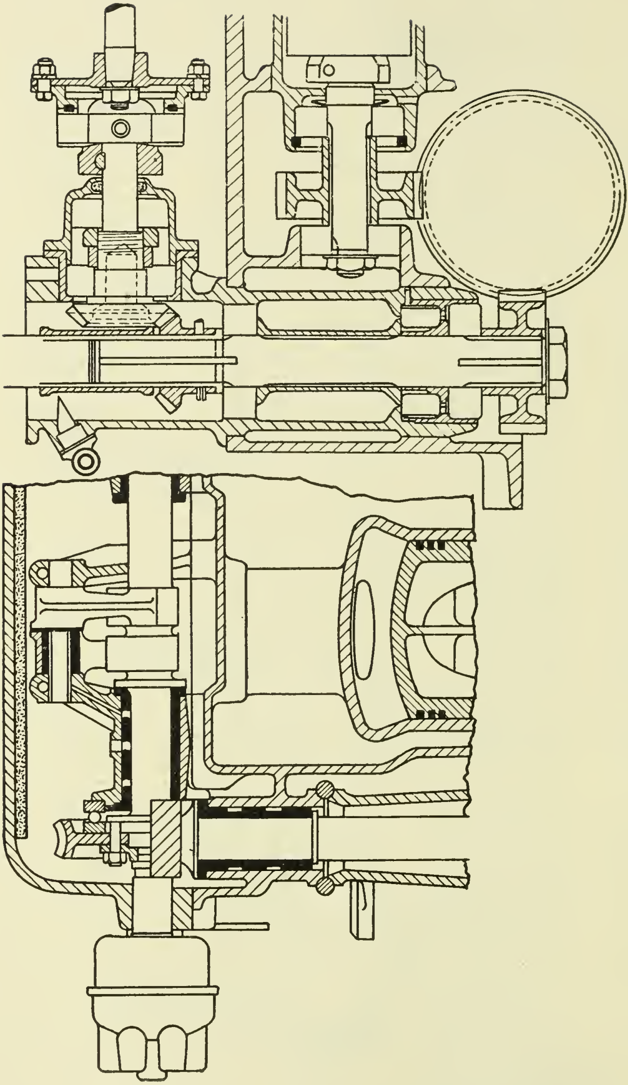

Fig. 4—Camshaft Drive of Lanchester Engine

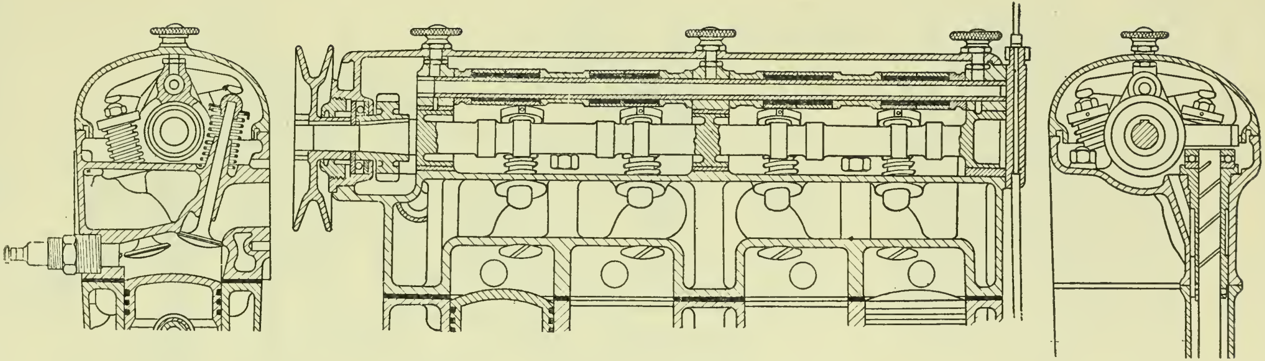

Fig. 5—Details of the Ansaldo Engine

A Cross-Section of the Upper Portion Is Presented at the Left and the Middle Drawing Is a Side View of the Same Portion. The View at the Right Gives Details of the Camshaft Drive is apparently provided for by washers under the flange of this hub.

Details of the camshaft drive on the Lanchester and Ansaldo engines are presented in Figs. 4 and 5.

SPIRAL BEVEL GEAR DRIVE

Probably the most prominent make of small car employing an overhead camshaft is the Wolseley. The construction which is shown in Figs. 6 and 7 is similar in some respects to that on the Napier engine but is simplified. The drive of the camshaft is by spiral bevel gear instead of worm gear. The spiral bevel gear drive is preceded by a silent chain drive from the crankshaft

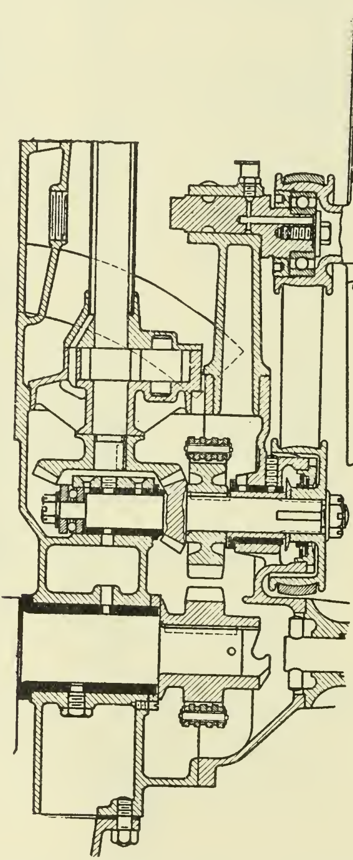

FIG. 6—OVERHEAD CAMSHAFT DRIVE ON THE WOLSELEY ENGINE

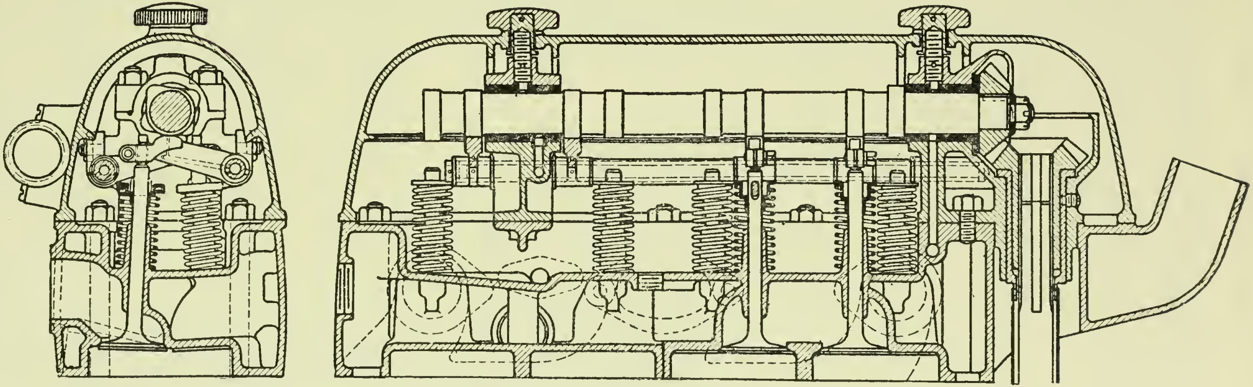

FIG. 7—THE UPPER PORTION OF THE WOLSELEY OVERHEAD CAMSHAFT ENGINE

to a parallel shaft directly above it, at a ratio of 1 to 1 and with a very short chain. The 2 to 1 reduction is obtained by the lower pair of bevel gears which permits the use of a small pair of gears at the top, thus making for neatness. It would be practically impossible to obtain this reduction at the bottom of the vertical shaft and still have the driving pinion on the crankshaft, for the reason that to clear the crankshaft bearing the pinion must be of large diameter, and this would necessitate such a large-diameter driven gear that the whole drive would become unduly bulky. As it is, the fan-drive pulley is mounted on a forward extension of the horizontal shaft and the gear-type oil-pump forms part of the mounting of the vertical shaft. The shafts of this drive are mounted in plain bearings exclusively, except for a ball thrust-bearing on the horizontal shaft. This is a small-bore four-cylinder engine with the camshaft supported in two bearings only. To reduce the unsupported length as much as possible, the two rearmost cams are arranged to overhang the rear bearing. The general arrangement of the valves, rocker-levers and cams is practically the same as on the Napier. Both the camshaft and rocker-lever shafts are supported by brackets on the cylinder-head, the rocker levers being located underneath the camshaft and giving a greater lift to the valves than what might be called the cam throw. Clearance adjustment is made by set screws in the ends of the levers which are locked by clamping bolts in the split ends of the levers.

The spiral bevel-gear drive seems to have much to recommend it for overhead camshaft engines, but it must be realized that in order that the gears may run together quietly and efficiently they must be adjusted accurately. As there are two pairs of gears four adjustments are required and provision must be made for any lack of alignment or relative motion due to differences in heat expansion between the upper and lower bearings of the vertical shaft.

This problem has been attacked by two engineers of Bentley Motors, Ltd., an English company, that has taken out several patents covering a number of points of design. Referring to the left-hand portion of Fig. 8, the vertical shaft is made in two parts, which are joined by a telescopic splined coupling. Each part can be accurately adjusted endwise with or in its casing. At the bottom this is accomplished by providing the casing with a flange, which, instead of being bolted directly to the crankcase, is provided with a threaded ring. This ring can be screwed up or down over the integral flange and with it forms a flange of adjustable thickness. At the top a bearing box *a* can be rotated slightly by inserting a bar through the hole closed by plug *b*, and by being rotated is adjusted up or down by means of the helical slots *c* and pin screws in the casing extending into them. The camshaft is held against endwise motion by two thrust bearings at the forward end, and can be adjusted endwise by simultaneously turning the two nuts at the opposite sides of these bearings. It will be noted that the upper part of the casing for the vertical shaft is integral with the camshaft casing while the lower part is integral with the crankcase and the intermediate part

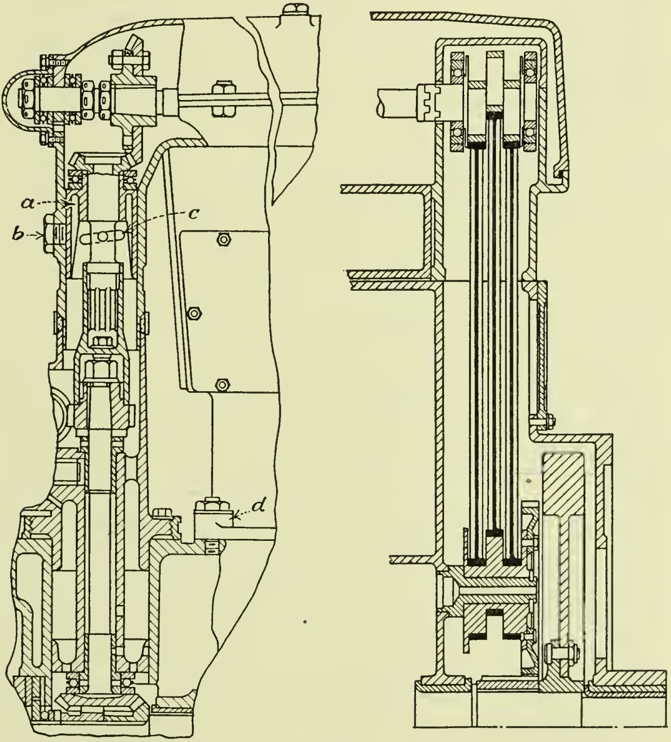

Fig. 8—At the Left the Camshaft Drive Used on the Bentley Engine and at the Right a Somewhat Unusual Form of Camshaft Drive That Is Used on the Leyland Straight-Eight Engine

bolts to this lower part. To permit the complete alignment of the upper and intermediate parts of the case the bolt-holes through the cylinder-base lugs *d* are made oblong. The endwise freedom of the connecting-rods on the piston pins permits of slight variations in the position of the cylinder-block on the crankcase. It is thus possible to adjust the mesh of both sets of gears and also the alignment of the vertical shaft housing parts.

Unusual Forms of Camshaft Drive

We now come to a number of rather radical designs of overhead camshaft drives and overhead valve-actuating mechanisms that have become known through recent patent publications. One of the cars that has attracted attention in the last year and a half is the Leyland eight-in-line. On the engine of this car an eccentric mechanism is used for transmitting motion from a half-speed shaft located close to the crankshaft and driven therefrom by a spur gear, to a short shaft normally in line with the camshaft. To obviate the difficulties due to the unequal expansion of the cylinder-block and the eccentric rods, the short shaft at the top has a support that is separate from the cylinder-head and is connected to the camshaft by an Oldham coupling which compensates for any slight misalignment between the driving and the driven members. The eccentrics are apparently set at angles of 120 deg. This drive ought to be commended from the standpoint of continuity. This construction is illustrated in Fig. 8.

A still more radical design of valve gear has recently been patented in England by the Fiat Co. Upon the upper part of the valve-stem is mounted a trunnion carrying two conical rollers. The trunnion is mounted between a shoulder on the valve-stem and a coiled spring that is backed up with a nut and lock nut at the top of the stem. Surrounding the valve-stem guide is a shell with opposite vertical slots through which the arms of the trunnion extend. On ball bearings upon this shell are mounted a pair of face cams upon the circumference of which are cut spur teeth. In the particular design shown in Fig. 9 there are four valves in the cylinder-head and all the cams are geared together. It is obvious that as the cams are revolved the valves are opened and closed positively, except for the slight play of the springs. A valve gear of this type should permit very high engine speeds, but the cost of manufacture is undoubtedly high.

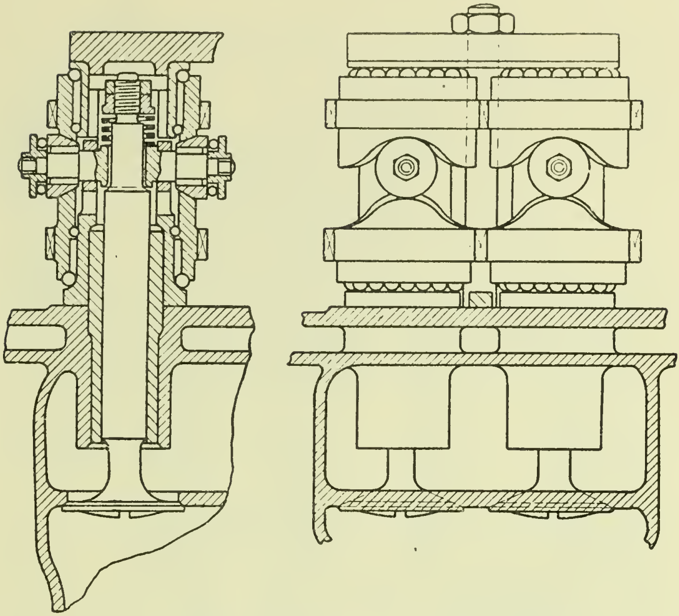

FIG. 9—OVERHEAD CAM GEAR ON THE FIAT ENGINE

VALVE MECHANISMS

At present three methods are in vogue for the operation of valves from overhead camshafts: The cams can act directly on cam followers secured to the end of the valve-stem, as in the Hispano-Suiza engine; single-armed levers or adjusting blades can be interposed between the cams and the valve stems; or the cams can act on the valves through the intermediary of tappet levers as in the Liberty aircraft engine. With the first-mentioned arrangement means for adjusting the valve clearance must be provided on the valve-stem itself. This problem was neatly solved by Birkigt, the designer of the Hispano engine. He drilled out the valve-stem and inserted a screw with a flat head with slotted rim. Underneath the screw head is a washer which has splines or keys on its inner circumference that engage into slots in the end of the drilled valve-stem, so that the washer can slide longitudinally on the valve-stem for a limited distance but cannot rotate with respect to it. The valve-spring presses the washer strongly against the screw head and the screw is thus locked. On the circumference

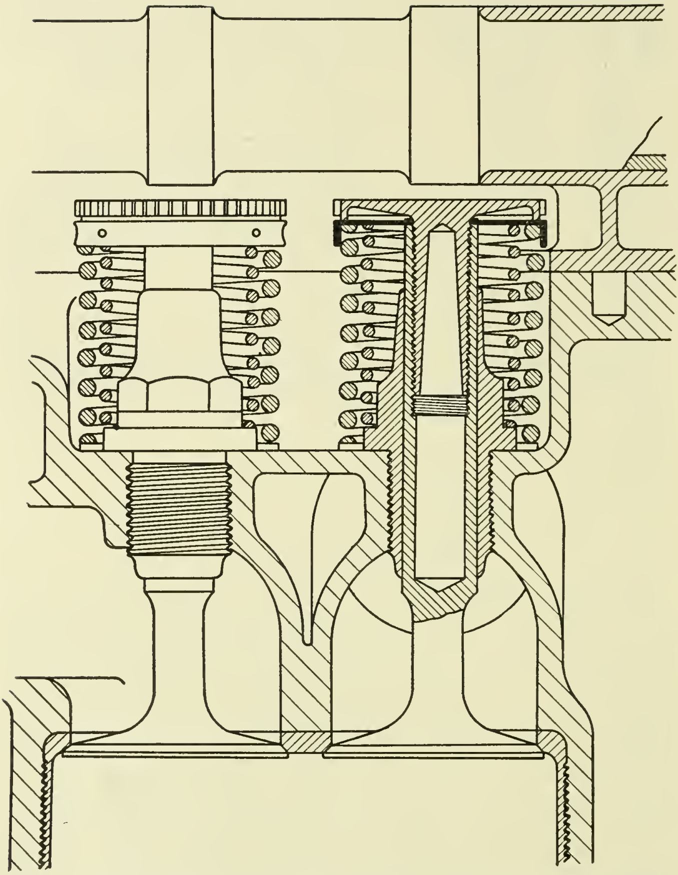

FIG. 10—ON THE HISPANO-SUIZA ENGINE A SPECIAL ARRANGEMENT FOR ADJUSTING THE VALVE CLEARANCE BY INSERTING A SCREW IN THE VALVE-STEM IS EMPLOYED

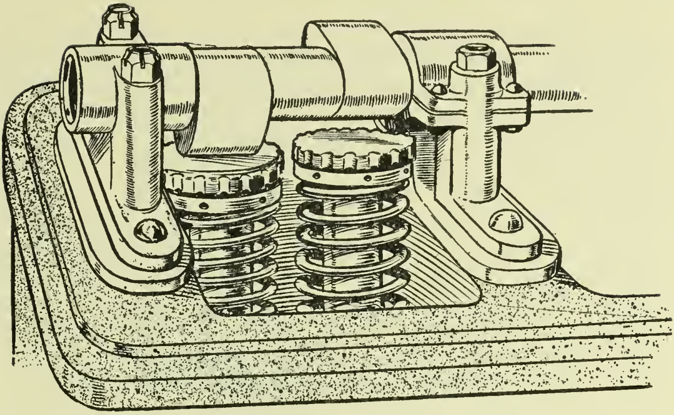

Fig. 11 — Sketch Showing the Relative Position of Valve Operating Cams That Act Directly on Followers Secured to the End of the Valve-Stem in the Hispano-Suiza Engine

of the washer there are several radial drill-holes. A pin wrench is provided to engage into the holes and slots of the washer and screw head and permit the adjusting screw to be screwed farther into or out of the valve-stem. The details of this arrangement for adjusting the valve clearance are apparent from an inspection of Fig. 10,

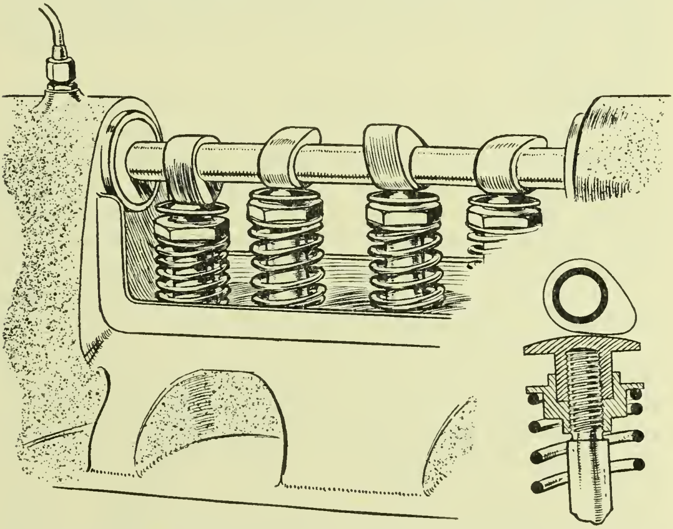

Fig. 12—Cam Mechanism That Was Used in the Dawson, a British Car That Is No Longer Built

while the relative positions of the cams and the followers on the end of the valve-stems are brought out in Fig. 11.

Fig. 12 shows the system applied to the British Dawson car which is now no longer built. As will be seen, it resembles the Hispano in being of the direct-acting type and has similar means for the adjustment of valve-clearance. Despite the prevalent idea that Hispano claims a master patent on direct operation, Dawson continued to turn out engines with this type of valve system without serious interference for about two years. The failure of the firm was not ascribable in any way to the type of valve-gear used on the engines. The scope of the Birkigt Hispano patent apparently covers only the particular means of adjustment, including the pin wrench.

It is, of course, entirely possible to arrange overhead-valve mechanisms in a way similar to that used with the ordinary L-head engine. That is, between the valve and the camshaft can be placed a push-rod which is provided with an adjusting nut or adjusting screw and lock nut. The objection to this construction is that it makes the engine abnormally high. An overhead camshaft engine is much higher than an L-head engine in any case and if a push-rod is placed between the valves and the camshaft the height of the engine is increased by the length of the push-rod. There may be instances where this is of no great consequence, as where a very small engine is placed under a normal-sized hood, but in most cases it would be objectionable.

Bugatti developed an overhead valve gear which overcame this objection. He uses curved quadrant-shaped push-rods with rollers at both ends. The cams on a camshaft placed centrally over the engine press against one of these rollers in a horizontal direction, while the other roller presses vertically down upon the valve-stem. The quadrant-shaped push-rods were located in guides bolted to the sides of the cylindrical camshaft housing with three bolts. The design illustrated in Fig. 13 is an early one, but the curved push-rod construction has been retained by Bugatti in a modified form. In this early design the provision for the adjustment of the clearance consisted apparently of the use of shims under the push-rod guide. Very frequently in recent designs the valves in the head are placed at an angle to the axis of the cylinder, one valve on each side of the central vertical plane of the engine, and the camshaft is then placed centrally over the cylinder-heads; an arrangement which

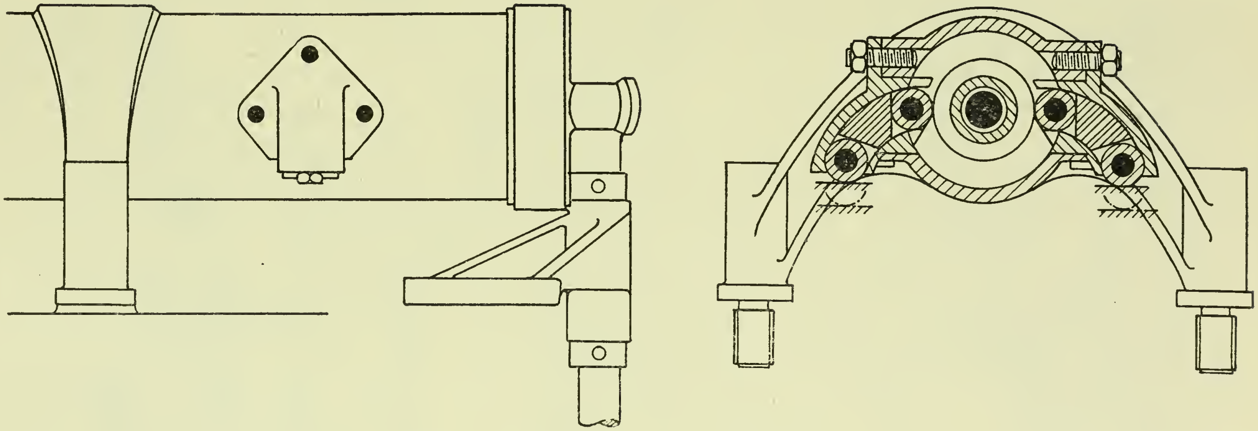

FIG. 13—THE BUGATTI OVERHEAD VALVE GEAR WHICH IS CHARACTERIZED BY CURVED QUADRANT-SHAPED PUSH-RODS WITH ROLLERS AT BOTH ENDS

makes for symmetry. On smaller engines the plan of setting the valves at an angle is not so practical, for it calls either for a wide cover or makes it necessary to leave the valve-springs unprotected and have the tappet-levers extend through the walls of the cover. In aircraft engines this latter practice is widely followed, because dust and dirt are seldom factors to be considered in that line and the additional noise also is of no importance. In passenger-car engines vertical valves seem to be preferred and to make it possible to operate these valves from a central camshaft single-armed instead of double-armed tappet levers are used. The two valves are sometimes placed on opposite sides of the longitudinal central plane of the engine and are offset in the fore-and-aft direction sufficiently so that the two levers do not interfere. This arrangement permits the use of comparatively long lever arms that reduce the side pressure on the valve-stems and the enclosure of the entire mechanism in a compact housing.

One of the simplest, in fact crudest, overhead camshaft applications is that of the Rhode, which is shown in Fig. 14. This is a British light car with a four-cylinder engine of 67-cu. in. capacity, selling at a low price. Its camshaft, which is driven by a train of helical pinions, actuates the valves through levers of rectangular stock pivoted on adjustable standards screwed into the

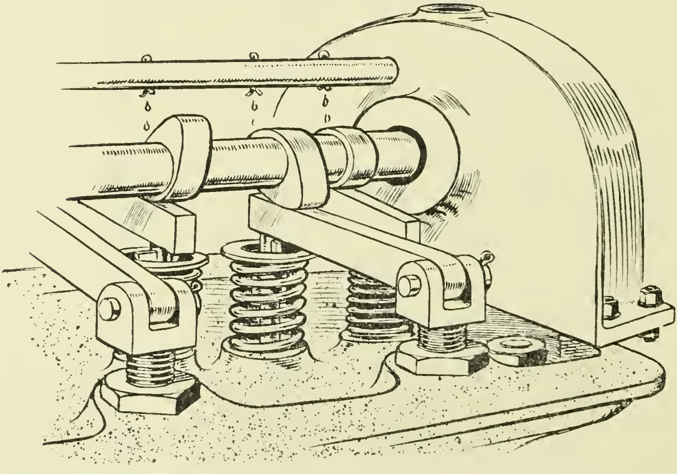

Fig. 14—Cam Mechanism on the Rhode, a British Light Car Having a Four-Cylinder Engine

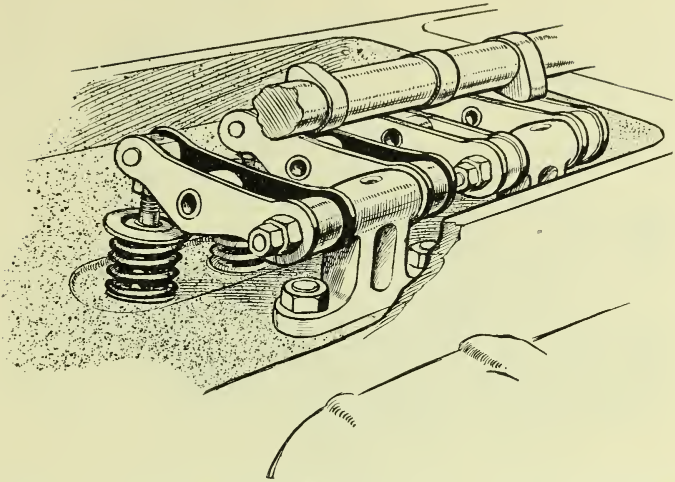

FIG. 15—VALVE OPERATING DETAILS OF THE 18-HP. FOUR-CYLINDER PHOENIX ENGINE

cylinder-head and provided with a lock nut. The camshaft is offset from the valve-stem and the clearance adjustment is made by varying the height of the pivot-pin centers. This implies first removing the pins and then raising or lowering the standards in half turns, a means of adjustment which is not very precise. As the drawing indicates, individual oil-drips from an oil pipe running from one to the other of the pressure-fed camshaft bearings are provided for the cams.

A British chassis of moderate price, with a four-cylinder engine of 180-cu. in. capacity having an overhead camshaft, is the 18-hp. Phoenix, of which the valve-operating details are shown in Fig. 15. This also has pivoted levers interposed between the valves and an offset camshaft, but the levers in this case are built up and somewhat resemble the links of a roller chain; in fact, the cam makes contact with a roller on the intermediate cross-pin. Provision for clearance adjustment consists of cap nuts on the valve-stems; the lubrication system includes pressure feed to the shaft bearings and separate drips onto the cams from a longitudinal pipe not shown in the illustration. The drive is by helical gears and a vertical shaft having a dog coupling that is provided with offset jaws to ensure correct reengagement within limits.

The Beardmore engine, illustrated in Fig. 16, also

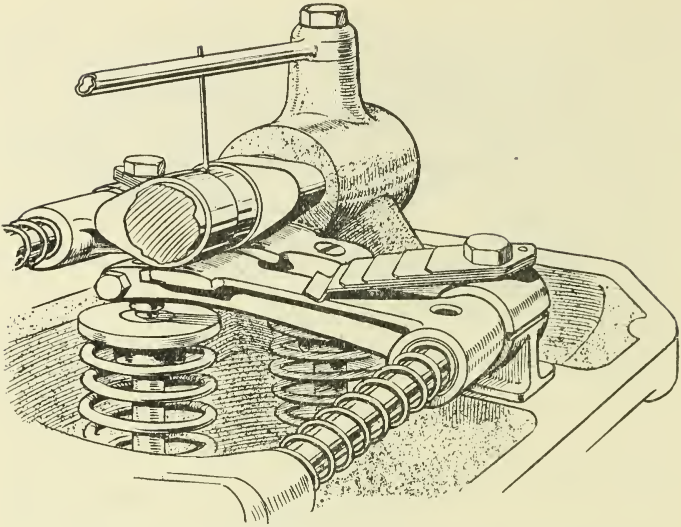

FIG. 16—THE BEARDMORE ENGINE THAT EMPLOYS PIVOTED LEVERS WITH AN OFFSET CAMSHAFT TO OPERATE THE VALVES

comes within the pivoted-lever class with an offset camshaft, though in this case the levers for inlet and exhaust-valves respectively are pivoted on opposite sides of the cylinder-heads. The lubrication details are similar to those of the Phoenix. Small leaf-springs prevent chattering of the levers. Pairs of levers are separated on their pivot shaft by helical springs that take up end play and allow the levers to be moved to one side to facilitate valve removal and clearance adjustment. The means for doing this consists of threaded studs in the lever ends that make contact with the valve-stems and are locked by a pinch bolt.

A design of distinct interest is that of the Diatto, which is shown in Fig. 17 at the left. This has pivoted and tapered tongues interposed between cams and valves, the camshaft center being immediately over the valve-stem centers. The clearance adjustment is effected by moving the tongue toward or away from the camshaft center-line, the horizontal supports being threaded, located in clearance holes and locked by nuts in the wall of the overhead valve-chamber as shown. Without being crude this is a simple design that should lend itself to economical production far better than many overhead camshaft arrangements. The angularity of the tongues must be large at

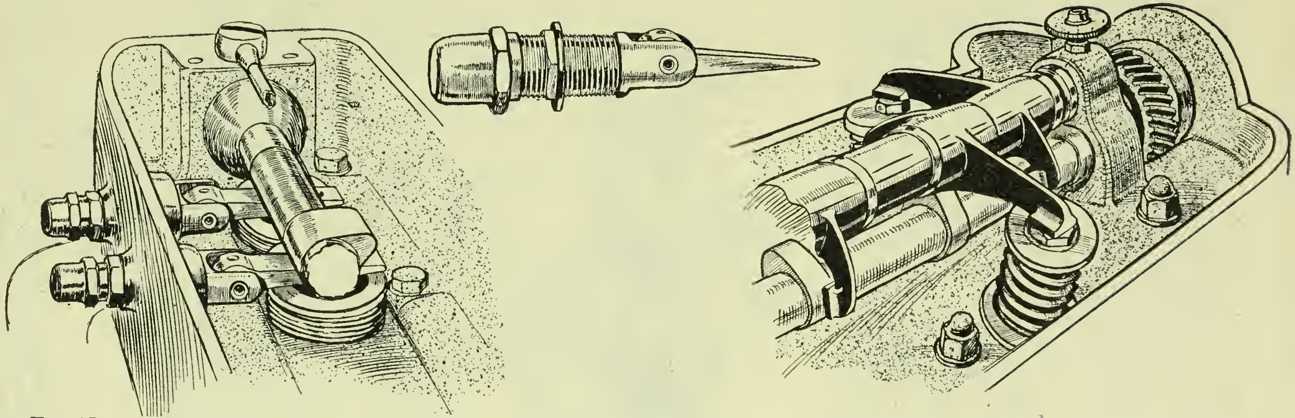

Fig. 17—The Diatto Cam Mechanism (at the Left) That Employs Pivoted and Tapered Tongues Interposed Between the Cams and the Valves and (at the Right) the Overhead Camshaft of the Ansaldo Engine That Uses Rocking Levers with Bearings on a Shaft Immediately Above and Running Parallel to the Camshaft

full valve-openings and would be liable to result in undue side thrust on the valve-stems, but there appears to be no valid reason why the tongues should not be longer and pivoted farther away from the valves to reduce the angularity and the side thrust. With efficient lubrication the wear of the tongues should not be excessive. In any case they could be made easily and cheaply renewable.

Typical of many other overhead camshaft designs, the Italian Ansaldo shown in the right half of Fig. 17 is in the class embodying rocking levers with bearings on a shaft immediately above and running parallel to the camshaft from end to end, with the valves on opposite sides of the cylinder-head. The valves are usually, as in the Ansaldo, outwardly inclined at a slight angle from the vertical. Although in the case illustrated the clearance

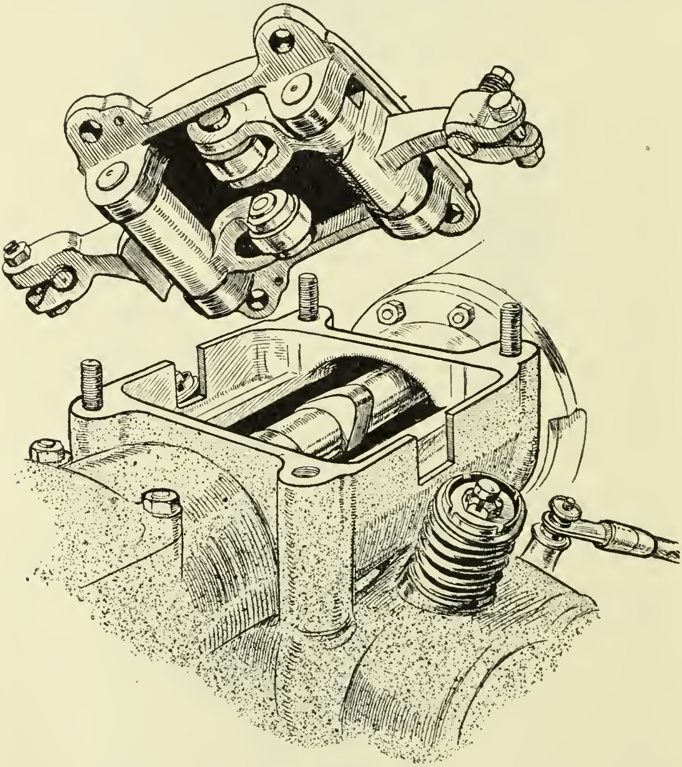

Fig. 18—The Overhead Cam Mechanism on the Straker-Squire Six-Cylinder Engine in Which the Valves on the Opposite Sides Are Outside the Camshaft Casing

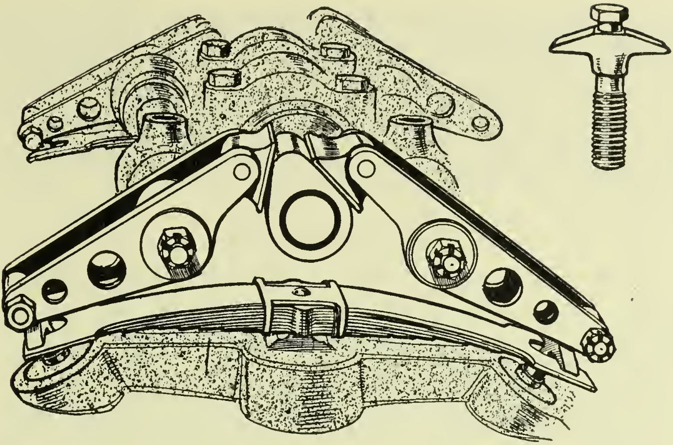

FIG. 19—THE OVERHEAD CAMSHAFT GEAR OF THE LEYLAND EIGHT-CYLINDER-IN-LINE ENGINE

adjustments are on the valve-stems, set screws and pinch bolts in the outer ends of the rockers are more in favor.

The Straker Squire arrangement, illustrated in Fig. 18, is unusual in two respects: In the first place it is used on a six-cylinder engine with separate cylinders; in the second place the valves on opposite sides are outside the camshaft casing. The drawing is practically self-explanatory, except that it does not show that the camshaft casing is a unit casting running from end to end of the cylinders and secured to them by the studs and nuts that hold in place the individual cover-plates that bear the roller-ended rockers. The drive in this case is by bevel gears at each end of a vertical shaft.

Still more unconventional is the Leyland eight-cylinder-in-line overhead camshaft gear, illustrated in Fig. 19, the drive for which already has been described. Only one cam is used for the inlet and exhaust-valves of each cylinder, the cam acting upon slipper followers pivoted to built-up rockers. These at their outer ends bear upon T-pieces screwed into the hollow valve-stems, while the springs are semi-elliptic laminated units forked at their ends to engage with and exert a lifting pressure on the T-pieces and through them upon the valves. With a total disregard for the cost of production, the Leyland at £10,500 for the chassis is the most expensive British car, the designer having evidently aimed at securing silent operation by endeavoring to prevent valve-spring rebound and so adopted the laminated type. The pivoted slipper followers may also have appeared to hold out possibilities in the same direction, but by being satisfied with the compromise valve-opening diagram that is obviously necessitated by using the same cam for inlet and exhaust, he would seem to have nullified any gain in efficiency that might have been produced by the overhead valves.

THE DISCUSSION

F. E. Moskovics:—Is not the chief difficulty in the direct application of the cam to the valve due to the thrust on the stem rather than to the clearance? I have had considerable experience in that work. I have had the benefit of investigating practically all of the racing cars of foreign make that come to Indianapolis for the 500-mile cup-race. That can be illustrated best by the lengths to which both the Peugeot and the Ballot designers have gone. They use a hardened cup that fits over the top of the stem, and that cup is in a guide which carries the thrust entirely free from the stem. Louis Chevrolet also uses a similar form.

I think there is no possible question that, from a technical standpoint, the overhead camshaft has certain advantages, but it is difficult to get a good drive to it. Chains cause trouble due to stretching. The bevel-gear thrust can be balanced but it is a pretty exact piece of work. All of these troubles are eliminated in the well-designed, well-built spur-gear drive with a side camshaft.

One point that Mr. Heldt's paper did not bring out strongly enough is the service problem. The passenger-car engine, be the design ever so good, does occasionally become carbonized. It must be demounted and mounted again. I submit that, placed in the hands of the ordinary small-town garage-man, the demounting and remounting and timing of an overhead-camshaft job is a vital problem in itself.

The location of the camshaft in the side of the engine has one other advantage in that it is in a large mass in the engine which you might say is an automatic muffler, for the noise of the camshaft is mingled with the other noises in the crankcase. I know of no place that is a better sounding-board for noises than the head of an engine that has a thin light cover. We have tried all sorts of things to deaden these noises, but it is just about one of the best drums for noise that there is. I believe the problem of the lubrication of the bearings of the rocker-arm type can be eliminated. I think almost every maker has succeeded in bringing oil up there now by using pressure; that is the general practice.

H. M. CRANE:—I will not attempt to speak for any particular type of engine, but a progressive history of three different types that I have been able to go through with myself is interesting. We first built some L-head six-cylinder engines of 4-in. bore and 5-in. stroke. Subsequently, we built push-rod overhead-valve engines of the same dimensions. During the last year I have been operating an overhead-camshaft engine of 4¼-in. bore by 5-in. stroke. I shall state briefly the relative qualities of those three engines. I think that the mean effective pressure can be made almost the same in all of them, but the L-head engine is much more prone to detonation, not of an order that possibly causes loss of power, but of an order that is very unpleasant to the driver and will cause him to retard the spark to prevent it and thus reduce the power. On the other hand, the L-head engine has a distinctly greater turbulence, which gives the advantage of a shorter spark-advance, the difference being that with the L-head engine a spark-advance of 25 deg. on the flywheel is ample for engine speeds up to 2400 r.p.m., while with the overhead-valve engines using two spark-plugs per cylinder from 30 to 35-deg. spark-advance is required to obtain the same results.

The interesting thing about the first overhead-valve engine that we built using push-rods is that it seemed to be quieter per se than a similar L-head engine. This may have been due to the form of the cylinder-head. In our experience the most difficult noise to contend with is caused by the seating of the valve. If it seats slightly in a diagonal way so that the stem slaps across, or if the seat has a sounding-board effect, the noise is very considerable. The opening of the valve, compared with this, is almost noiseless. The overhead-valve cylinder-head construction, with its heavy water-jacketing and stiff ribbing, seems to help the deadening of the exhaust-valve sound.

In our push-rod job we used cylinder-heads in two sets of three each, which of course is very easy to do in an engine of that type. With the overhead-camshaft job it was obvious that the only really good way is to use a single cylinder-head, and there we find the chief advantage of the overhead-camshaft job as a practical thing.

Mr. Heldt speaks of this in his paper and I think it is an excellent way of deciding on an engine; that is, which is the easier to take care of.

The overhead-camshaft engine cylinder-head on this rather large engine weighed about 135 lb. That is not a one-man job to dismount. Evidently it needs at least two men, and it is apt to require a chain-hoist. On the other hand, the heads of the cast-in-three cylinders, using rocker-arms and push-rods, weighed about 45 lb. each. It is a very easy one-man job to dismount. It is also easy to handle on the bench for grinding valves and has every advantage.

Our experience on cylinder expansion on this engine, which of course is of a moderately short stroke, indicated that it was wholly possible to figure on not over a 0.01-in. maximum variation on the inlet-valve side, if any means were taken to see that the push-rods are warmed proportionately to the cylinders to some extent. We have found again and again that it is possible in modern cam design to provide for such a variation in expansion without affecting the rate of opening and closing. I think this is the type of design that is called using a ramp on the cam. The principle is to determine a range of angle on the cam through which the speed of action of the lifter is the same, and if that is done it is obvious that no matter at what point the cam takes hold, the shock of the blow or of the valve seating will be the same. We have used that arrangement on exhaust-valves and found no difficulty at all from timing variations.

When you add to the lack of accessibility of the overhead-camshaft engine the difficulty of making the drive quiet, in the first place, and keeping it quiet afterward, I cannot see that there is any chance for argument at all in the passenger car. That is partly because I have always been an advocate of slow-speed engines, and still am. I think that the passenger-car engine of the most efficient type is one that reaches its peak at about 2400 r.p.m. or possibly 2500 r.p.m., and any attempts to make a quiet engine capable of running very much faster will meet with very little success; in fact, if you wish a very high-speed engine and want to have it quiet, I am inclined to think that the L-head engine is the most suitable form. I believe that you can make the parts for suitable bearing areas lighter in an L-head engine than you can in any form of overhead-valve engine. With a moderate speed for the engine it is possible to use large bearing areas with large oil-films and, when that is done, I have no hesitation in saying that an overhead-valve engine can be made that is to all intents and purposes absolutely silent so far as the valve operation is concerned, but it is bound to be the result of a strenuous course of development.

The difficulty of predicting what the result will be with any form of cast aluminum and cast-iron and forged metal construction, as to its acoustic or sounding-board effects, is such that we do not know anything about it. We all know that the Cadillac engine is surprisingly quiet, considering the excessive pressure of valve-springs used and the fairly good weight of the parts. Examination of the design does not give any hope on the face of it that it should be quiet. The valve-operating mechanism is mounted on an excellent sounding-board, apparently, and there is nothing else there to give one the idea that it should be quiet; but I think actually it is extremely quiet in view of what is being done by the valve-operating mechanism. On the other hand, a design with very much lighter springs, lighter valves possibly, and smaller parts all the way through, may cause excessive noise because the parts happen to line up in about the form of a telegraph sounder that brings the full amount of noise out of any mechanical slackness.

J. G. VINCENT:—I wish to second everything Mr. Crane has said in regard to the characteristics of the three types of engine. It happens that I went through just about the same procedure about 2 years ago in experimental work on the straight L-head, the push-rod-operated overhead-valve and the straight overhead-camshaft engines.

Taking the overhead-camshaft job first, it seems to me that the advantages and disadvantages of this type of engine have been very well brought out. I think that lack of accessibility is the most important element working against this engine, and I believe that we must all take into consideration the service end of the business. Granting that you can make a quiet drive to the camshaft, there is one other point that I think is important. It is that the tappets and all the operating parts are right on top of the engine, where they certainly are easier to hear than when they are located down in the crankcase.

In regard to the matter of losing the timing when you demount the engine, I think that is very important. In connection with the better location of the camshaft in the crankcase for lubrication, I think it gives the crankcase position unquestionable advantages, particularly as that is the easy place in which to drive the camshaft.

When you come to the push-rod-operated job, so far as drive is concerned, it is exactly the same thing as the L-head engine; whether you choose to use spiral gears of some kind or a chain drive is another matter. Either job can be made good, but I believe that the chain gives better results from the manufacturing point of view.

So it seems to me that, as Mr. Crane pointed out, it is clear that it comes down to a practical point of view where any quantity of production is involved as between the L-head and the push-rod-operated overhead-valve engines. Undoubtedly, both those engines will be developed and used to a large extent, because they both have some advantages. Perhaps they can be made equally quiet. Of course, the L-head engine with the demountable head still has some advantage from an accessibility point of view, and the question I would like to raise is: Is it worth while? In other words, what do you really gain by putting the valves in the head? I have been unable to determine any advantage, and I think there are many other things about the engine that can be worked on which may give as much or more advantage. For instance, very careful attention to the carburetion will accomplish wonderful results. When you come to consider that the passenger-car engine is operating most of the time at a very small percentage of its maximum output, you will find that with the compressions you can use there is mighty little to be gained in either power or economy with the overhead type of engine. I was not able to obtain anything in excess of a 1-per cent advantage; so, it is very questionable in my mind, because the overhead valves add some weight to the engine, whether the small increase in efficiency will make up for the added weight you have to put in the engine. The best way to get economy is to reduce the weight and, as Mr. Crane said, use an engine of slower speed with a relatively moderate gear-ratio. It will be interesting to see which one of these two types of engine will succeed in the long run. Probably both will always be used. So far as I can see, however, the overhead camshaft cannot be manufactured in quantities on account of the cost and the difficulty of making the job quiet.

F. S. Duesenberg:—In regard to the setting or the timing of the gears in the service-station, I believe that as we are building our engine at present there is less liability of getting the timing off than in any other type of engine, because we use the upper gears mounted in the head and on a vertical shaft that is driven by one jaw in such a manner that it cannot be put together wrong. If you take the head off, there is a possibility of turning the crankshaft over one complete turn, and in that way the ignition might have to be set but, so far as valve-timing is concerned, it is impossible to get that wrong. I really feel that there is not any disadvantage in this point. The quietness, of course, is something that needs to be considered, and I believe it is easy to make a rocker-arm or the overhead camshaft quiet, but of course the gear noises must be taken care of. I have been driving a Willys car for the last year and I bought it to test out the gears. When I first got the car it was very noisy, but no adjustments have been made and, after 10,000 miles, it has surprised me on account of absolutely losing all the noise it had when I first got the car. For this reason I feel that it will not be an impossibility to get a very quiet gear if it is properly made. However, I do feel that any gear that will be quiet after a few thousand miles of use is apt to be slightly noisy at the beginning, because it has to be set up very close.

The accessibility of an overhead camshaft I really feel is rather better than it is with the other type of construction; it leaves the crankcase very free. In the arrangement that we have with the overhead camshaft, the gaskets bothered us at first on account of the variation in their thickness, but that was easily overcome and we have experienced no other trouble. We find it possible to take the present type of head off, grind the valves and have it back on the job in 2 hr. Two men can do that easily. The weight of the head is to be considered and does not make it as accessible or as easy as some of the other forms; our head weighs 140 lb., complete.

I might cite an experience that Bill Murphy had in a race at Los Angeles about a year ago where they had four heats in the final. Murphy burned a valve in the first heat; he tried to run the second heat and found his carburetion was unsatisfactory, so he stopped and took the head off. He had a new head without a camshaft; so, he removed the camshaft from the old head, put it into the new one, put the head back on and had the engine running in 32 min. from the time he stopped. In the next heat, he broke the world's record by 3 m.p.h. This incident shows that those things can be made very accessible and very easy to handle. I feel that the combustion-chamber on an overhead-camshaft engine can be made so clean and so easily that it really is a good production job, but I think that a little greater accuracy is necessary if you use the gear drive than if a chain drive or the ordinary L-head is used.

R. ABELL:—Mr. Duesenberg's experience and my own have been about the same in regard to the overhead-camshaft engine; namely, that the dismounting of the head and valve mechanism and its replacement is a shorter job than with any other type of engine. With a helper, I have taken the head off of my engine, removed the camshaft, inspected the valves, reassembled the head, timed and adjusted the valves and had the engine in operation again in less than 1 hr. The principal difficulty has been with gear noise, but it has been amply proved in quantity production that this feature has been overcome and that overhead camshafts are now remarkably silent. I think the important thing in the design of the overhead-camshaft job is to make the adjustment very accessible; in other words, to have a cover-plate removable from the front so as to expose the entire drive in order that fine adjustments can be made easily, with good opportunity to examine the meshing and lost motion of the gears. If the noise is more than is desirable with a new set of gears, a slight amount of use will indicate the adjustment required and, as adjustment is permitted with this type of drive, it is important to have the adjustments as accessible as it is possible to make them.

In most overhead-camshaft engines, to make adjustments of the lower gears it has been necessary to disassemble the entire engine and then work in the dark; so, in an economical production design, it is very important to provide for accessible adjustment at this point.

R. E. FIELDER:—I have listened with great interest to Mr. Heldt's paper. Certainly its discussion is most illuminating. The point that strikes me most forcefully is the fact that, even at this rather late date, the poppet valve and its method of operation are still far from perfection. As a matter of fact, there seems to be a very general disagreement in regard to certain controlling design factors.

Nearly all the speakers have referred to certain disadvantages that are inherent in the various types of poppet-valve construction, and one instinctively wonders why it is that designers tolerate the disadvantages that they have mentioned, the most important of which are noise, inaccessibility and difficulty of assembly and of adjustment. Certainly, these disadvantages are not found with the sleeve-valve type, the pocketless combustion-chamber, simplified cylinder construction and accessibility of which stand out so prominently.

D. B. WEBSTER:—One of the cases cited in Mr. Heldt's paper is that the Leyland eight-cylinder-in-line engine is designed with a camshaft operated by connecting-rods. That seems to me to offer a way out of the gear noise. Has anyone had any experience with that type of operation? Are there some objections to it that we have not heard of?

MR. DUESENBERG:—The rod type of control shown on the Leyland engine has been experimented with by several people. It was tried out a few years ago by one of the Western engine makers and was found absolutely unsuitable for use with a separate head construction. The change of the gasket or the variation that occurred in the engine when it was taken down was such that the distance in length would change and the rods could not be changed, and there was no way to get them adjusted properly so that they could be operated satisfactorily.

P. M. HELDT:—That is being taken care of in the Leyland engine by using a short shaft. The connecting-rods do not drive directly on the camshaft, but to a shaft that has an Oldham coupling connected with the camshaft.

A. L. NELSON:—With regard to comparing the L-head and the valve-in-head engines, I think most of us will agree that very fine results have been obtained by both types so far as present-day practice is concerned. It is my belief that the requirements of today must be set aside in the light of the great advance that has already been made in the way of providing fuel "dope" that makes practicable the use of compression-ratios of about 7 to 1. To obtain an appropriate combustion-chamber on an L-head engine having a high compression-ratio, together with reasonably large valves, is an exceedingly difficult matter when due attention is given to port areas that permit the attainment of a fairly good volumetric efficiency. We have observed very often an actual loss of power in experiments when the compression of L-head engines was increased. This we know is contrary to what should be expected. Invariably it is found that the charge is throttled on entering the cylinder. This throttling of the charge accounts for the lack of increased power.

In the case of the valve-in-head engine, the compression-ratio can be increased readily without throttling the charge. It is thought that designers should bear this important consideration in mind when looking into requirements for the near future. In short, the valve disposition that is universally approved in aviation-engine work should eventually become the universal practice in the high-efficiency engines that we will be called upon to produce in the near future. There appears to be no room for doubt that the highest economy, with the least amount of detonation or fuel knock, will be obtained by adopting valve-in-head engines.

MR. HELDT:—I can say only that there seems not to be very much sympathy for the overhead-camshaft engine among the scientists here. I have followed foreign periodicals pretty closely and have noticed that very many patents have been taken out on different constructions. If the overhead-camshaft engine is adopted here, I think it will be for the higher grade of cars where the cost of construction is not so very important. I believe it is not the proper kind of construction to be incorporated in a low-priced car.

A. C. WOODBURY:—A simple way to secure different periods of valve-opening from the same cam where a roller cam-follower is used is by varying the size of the roller. As an illustration, let us assume that it is desired to hold the exhaust and the inlet-valves open through periods of 230 and 215 deg. respectively, 115 and 107½ deg. of camshaft motion, and that the valve-stem clearance is to be 0.020 in. With a straight-sided cam, a 1¼-in. base-circle and a ⅞-in.-diameter roller for the inlet-valve, the angle between the two sides of the cam would be about 67 deg. On the same cam a 2-in.-diameter roller would produce the 115-deg. opening period required for the exhaust-valve, with the same lift as the inlet-valve.

So far as valve-timing is concerned, the slipper of the Leyland engine would act the same as a roller and the opening periods can be varied by changing the distance from the pivot-pin to the slipper face, but from Fig. 19 of Mr. Heldt's paper it appears that there is no such difference between these dimensions on the two slipper-blocks as would be required to make the customary difference between the two valve-opening periods.

VALVE ACTIONS IN RELATION TO ENGINE DESIGN[1]

By Chester S Ricker[2] and John C Moore[3]

The authors present and discuss the results obtained from combined road and laboratory tests made to determine the amount of power required to maintain a given car speed. The specifications of the car and its engine are stated and the variable-ratio rocker-arm of the engine is illustrated and its advantages explained, together with those of the valve-timing. The subject of manifold gas-velocity is treated in some detail, inclusive of a diagram showing the hot-spot or vaporizing device that was used.

The test data are reduced to curve form, eight charts being shown. The curves include those for brake horsepower, indicated horsepower, comparative performance, performance at different throttle-openings and at different loads, fuel consumption and indicated thermal efficiencies.

The first tests were made to determine the road resistance offered by a car running at a constant speed over a ½-mile course. This was the graveled Milton-Pike road, north of Connersville, Ind. The average outdoor temperature was 45 deg. fahr. and the barometer read 29.4 in. To determine the same points in the laboratory later with the engine and carbureter, graduated cards were fastened on the carbureter and on the ignition control so that we knew the exact point at which each of these instruments was set at each speed at which the car was run. The purpose of these cards was to enable us to duplicate the speed test on the dynamometer in the laboratory and determine what horsepower was required to maintain the car at each given speed. Another purpose of the tests was to make the data which we obtained comparable with the data A. L. Nelson obtained in 1920. The curves shown were plotted from the data obtained during these tests. The Lexington car with which the tests were conducted and the Ansted engine used in it had the specifications shown in Tables 1 to 5.

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[1] Mid-West Section paper.

[2] M.S.A.E.—Consulting engineer, Indianapolis.

[3] M.S.A.E.—Chief engineer, Lexington Motor Co., Connersville, Ind.

The type of rocker-arm shown in Fig. 1 is interesting; it has a variable ratio. It opens the valve slowly and then, as the valve continues to open, the ratio on the rocker-arm changes and, when the valve is wide-open, the end-point gives the rocker-arm nearly three times the ratio that it has at the start. In other words, the ratio

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TABLE 1—SPECIFICATIONS OF LEXINGTON MODEL ST CAR

Weight with Driver, Spare Wheel, Tools and Tanks Full, lb.	3,405
Weight, Less That of 159-lb. Driver, lb.	3,246
Allweather-Tread Cord Tire, Size, in.	32 x 4
Average Roll of Rear Wheel, in.	101
Number of Wheel Revolutions per Mile	628
Number of Engine Revolutions per Mile	2,908
Exhaust Cut-Out	Open
Rear-Axle Gear-Ratio	4⅝ to 1

TABLE 2—GENERAL SPECIFICATIONS OF THE ENGINE USED

Type	Valve-in-head
Number of Cylinders	6
Bore, in.	3¼
Stroke, in.	4½
Piston Displacement, cu. in.	224
Number of Crankshaft Bearings	3
Length of Front Main Bearing, in.	2½
Diameter of Front Main Bearing, in.	1¾
Length of Center Main Bearing, in.	2⅝
Diameter of Center Main Bearing, in.	2¼
Length of Rear Main Bearing, in.	3⅜
Diameter of Rear Main Bearing, in.	2⅜
Length of Crankpin, in.	1½
Diameter of Crankpin, in.	2¼
Lubrication System	Full Pressure
Lubrication Regulation	Vacuum
Cooling Regulation	Rayfield Thermostat
Carbureter Type	Rayfield Model M Horizontal
Carbureter Size, in.	1½

TABLE 3—AVERAGE WEIGHT OF 10 SAMPLES EACH OF MOVING PARTS TAKEN AT RANDOM

	Oz.
Cast-Iron Piston	24.32
Piston-Ring	0.85
Piston-Rings per Piston	1.70
Piston-Pin	3.44
Connecting-Rod Assembly, Small End	7.90
Connecting-Rod Assembly, Total Weight	45.47
Valve Alone	3.31
Valve Push-Rod with End	3.29
Valve Lifter, Roller and Pin	6.23
Valve Rocker-Arm Complete	4.34
Valve-Spring Retainer and Key	0.82

TABLE 4—PISTON AND CONNECTING-ROD SPECIFICATIONS

Piston Material	Cast Iron
Length of Piston, in.	3 17/32
Number of Piston-Rings	2
Width of Piston-Rings, in.	3/16
Length of Piston-Pin, in.	2 7/8
Outside Diameter of Piston-Pin, in.	7/8
Inside Diameter of Piston-Pin, in.	11/16
Connecting-Rod Section	I-Beam
Length of Connecting-Rod, in.	8 1/2
Number of Connecting-Rod Cap-Bolts	2
Diameter of Connecting-Rod Cap-Bolts, in.	7/16

TABLE 5—VALVE-GEAR SPECIFICATIONS

Diameter of Valves, in.	1.7500
Clear Opening of Valve Port, in.	1.6250
Diameter of Valve-Stem, in.	0.4120
Length of Valve, in.	5.6870
Combined Pressure of Dual Valve-Springs, lb. per sq. in.	75
Valve Rocker-Arms	Moore Type
Valve Rocker-Arm Ratio, at Opening	1.265 to 1
Valve Rocker-Arm Ratio, Full Open	3.047 to 1
Diameter of Valve Push-Rods, in.	0.2500
Diameter of Camshaft, in.	1.1250
Radius of Cam Base-Circle, in.	0.6250
Radius of Cam Point, in.	0.8437
Lift of Cam, in.	0.2187
Diameter of Valve-Lifter Rollers, in.	1.1250
Intake-Valve opens	10 deg. late
Intake-Valve closes	56 deg. late
Exhaust-Valve opens	50 deg. early
Exhaust-Valve closes	10 deg. late

is approximately 1¼ to 1 when the valve is cracked open, but when the valve is wide-open, the ratio is almost 3 1/16 to 1. This brings up an interesting point in connection with the rocker-arm for the exhaust-valve, because this valve opens against the full pressure in the combustion-chamber. This gives the maximum leverage or about a 1¼ to 1 ratio when the valve is cracked open; after that, the pressure being equal on both sides of the valve, the valve-gear has less work to do and this becomes less important.

Another point in connection with this variable-ratio rocker-arm is that a rocker-arm on the top of an engine rises and falls with the expansion of the cylinder-block. The push-rod may expand the same amount as the cylinder-block does or it may not. If the push-rod does not expand the same amount as the cylinder-block and the valves are adjusted when the engine is hot so that they have very little clearance, there is the possibility of

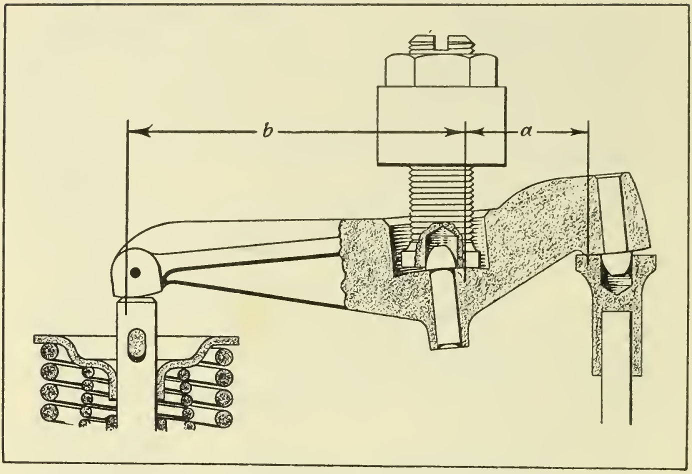

FIG. 1—VARIABLE-RATIO ROCKER-ARM
The Ratio of *a* to *b* Is 1:3 1/16

holding the valves open when the engine is cool, due to the variation in the push-rod and the cylinder shrinkage. That is clear to anyone who has had anything to do with an aluminum-cylinder engine. In fact, it has been found necessary to allow from 0.013 to 0.015-in. clearance between the rocker-arms and the valve-stems on an aluminum-cylinder engine when they were hot, to insure that the valves close when the engine is cold. Here we have a very small ratio, practically a 1 to 1 ratio on the valve-gear at the closed point; hence, this design is not so susceptible to variations in the length of the cylinder-block or the length of the push-rod, due to manufacturing difficulties or to temperature. The subject of the velocity of opening and closing the valve is likewise noteworthy. The rocker-arm starts off with the 1¼ to 1 ratio and ends with a wide-open valve having a 3 1/16 to 1 ratio. That undoubtedly affects the valve action. In fact, we know it affects the valve very noticeably in regard to quietness.

The valve timing of the Ansted engine is such that the exhaust closes and the intake-valve opens at 10 deg. past top dead-center; the intake closes 56 deg. late, while the exhaust-valve opens 50 deg. early.

The intake-valve of the engine mentioned by A. L. Nelson in his paper entitled Fuel Problem in Relation to Engineering Viewpoint[4] opens at 4 deg. past top dead-center and closes 60 deg. late, or only 4 deg. later than the intake-valve of the Ansted engine; hence, so far as the intake-valve timing is concerned, both engines are practically the same. The exhaust-valve of Mr. Nelson's engine opens 52 deg. before bottom dead-center.

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[4] See TRANSACTIONS, vol. 16, part 1, p. 325.

For convenience in comparing the speed of the car and the revolutions per minute of the engine, all the charts are made on both a miles-per-hour and a revolutions-per-minute basis, as follows:

Car Speed, m.p.h.	Engine Speed, r.p.m.
10	484
20	968
30	1,454
40	1,936
50	2,420
60	2,908

For each 10-m.p.h. increase of the car speed, there is about a 500-r.p.m. increase in the engine speed, and this is an easy way to judge the curves.

MANIFOLD GAS VELOCITY

In computing the velocities of gas in the manifolds of a high-speed engine, some problems develop. A sectional view through two manifolds is shown in Fig. 2; the upper one represents the manifolds that A. L. Nelson uses in his engine and the lower one a cross-section of the Ansted-engine manifold. In both engines the carbureter opening is given as *a* and the area of the longitudinal manifold in the cylinder-head as *b*. The feature in the comparison of the two engines and their performance

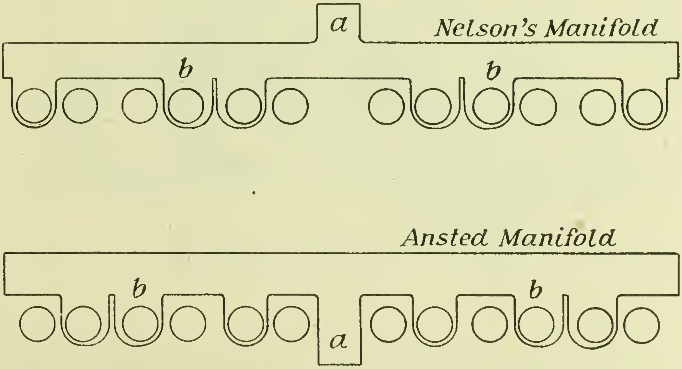

FIG. 2—COMPARISON OF THE MANIFOLD SECTIONS OF TWO HIGH-SPEED ENGINES, SHOWING THE DIFFERENCE IN THE DISPOSITION OF THE CARBURETER AND VALVE PORTS

is that the area at *a* in Mr. Nelson's manifold is large enough to maintain an average gas-velocity of 181 ft. per sec. when the engine is running at 3000 r.p.m. Apparently, he assumes in his design that only one-half of the gas comes over in this manifold, and therefore he gets a velocity of 175 ft. per sec. for one-half of the manifold; as the section *b* is reduced to 1½ in. from the section *a* which is 2 3/16 in. in diameter. These figures are computed for an engine of 295-cu. in. displacement. One difference exists between the two engines; the performance must prove whether it is for better or worse. In the Ansted engine, the manifold is 1¾ in. in diameter. The assumption made in its design was that more or less gas is handled over the entire manifold; so, instead of reducing this to maintain a uniform velocity for one-half of the gas, as Mr. Nelson assumes, it has been maintained at the same size. This difference is a manufacturing proposition. We question whether it was scrutinized

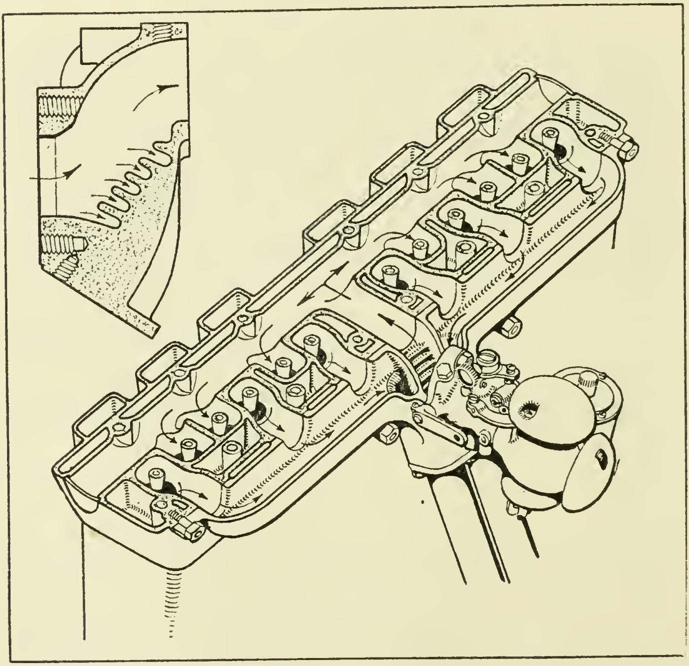

FIG. 3—CROSS-SECTIONS OF THE CYLINDER-HEAD AND THE HOT-SPOT GRID

The Exhaust Gases Pass along Either Side of the Hot Spot

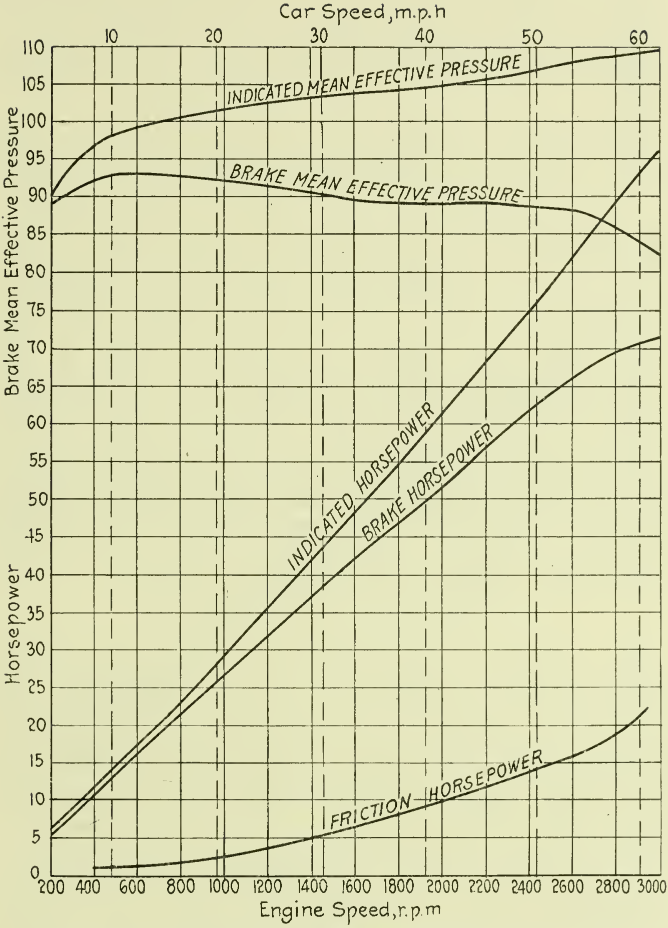

Fig. 4—Curves Showing the Brake Horsepower and Mean Effective Pressure Developed

very closely when the original design was made, but it worked so well that it has never been changed. With this 224-cu. in. engine we find by computation on the same basis that Mr. Nelson used that the gas velocities are very different. We find that the gas velocity at section *b* in this manifold is 97 ft. per sec., although the gas velocity in the manifold is 194 ft. per sec. at the same speed.

This is not so much a matter of comparison as of inducing thought along the lines of what size of manifold should be used in a six-cylinder head of this design. Some of the carbureter men may have some comments to make in regard to those figures. The gas-velocity figures are for an engine speed of 3000 r.p.m.; at 400 r.p.m. in the low range, the results are very startling. For example, on the Nelson engine, the gas velocity is reduced to 47.9 ft. per sec. through section *a* at 400 r.p.m., with 51 ft. per sec. at section *b*, while on the Ansted engine the velocity in section *b* of the manifold drops to 25 ft. per sec. where the gas-velocity in Mr. Nelson's engine is maintained at 46 ft. per sec. A drop of 25 ft. per sec. ought to throw the unvaporized gasoline down into this manifold and cause loading, but apparently it does not. We believe that this is due to the character of hot-spot or vaporizing device that is used at section *a*, a diagram being shown in Fig. 3.

INTERPRETATION OF TEST DATA

Fig. 4 shows the brake horsepower obtained with the Ansted 224-cu. in. engine. It will be noticed that the brake mean effective pressure on this engine, which was calculated from this brake-horsepower curve, is unusually flat for an engine of this type. It starts at above 90 lb. per sq. in. mean effective pressure and holds it out to about 2000 r.p.m., or about 40 m.p.h. The indicated mean effective pressure is taken from the indicated-horsepower curve that was obtained by the addition of the friction-horsepower values to the brake horsepower. We get an indicated mean effective pressure of about 100 to 109 lb. per sq. in. from the indicated-horsepower curve. This brake-horsepower curve was made from an experimental engine that was thoroughly limbered up, but just after we finished the test we had the misfortune to break a valve; this necessitated the use of a new unlimbered engine taken out of ordinary production stock to obtain the friction-horsepower curve that is shown. Hence this fric-

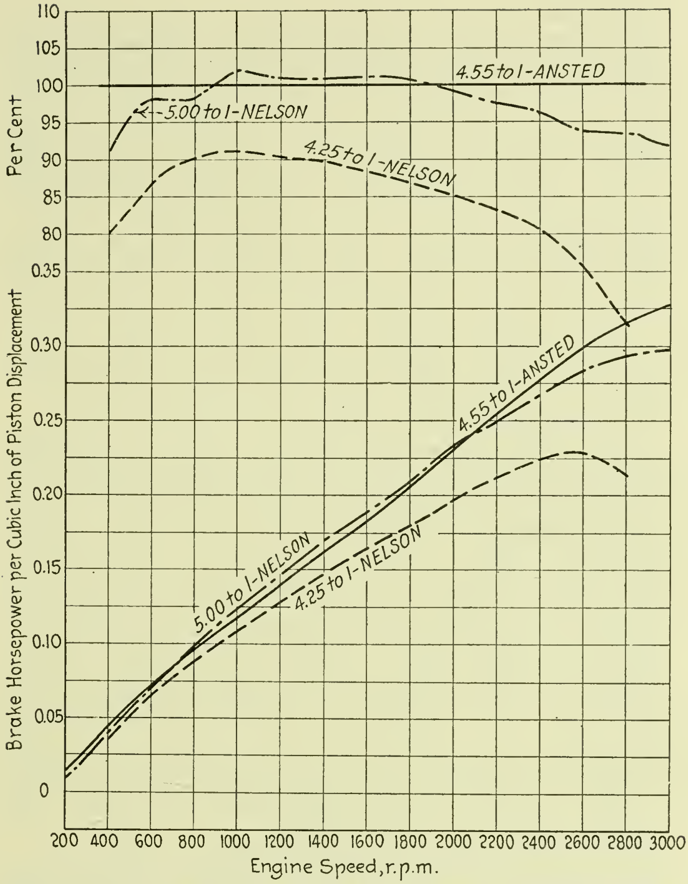

FIG. 5—COMPARISON OF THE POWER OBTAINED IN TWO HIGH-SPEED ENGINES

tion horsepower is probably much higher than the friction horsepower on a well limbered engine, but we gave the freer engine the higher-friction curve to be sure that the engine received no credit for something that it did not deserve.

In the second series of curves shown in Fig. 5, we tried to interpret the data we obtained from the brake-horsepower curve so that it would be comparable with the data in Mr. Nelson's paper. He gave the performance of a 295.2-cu. in. engine with a 4¼ to 1 compression-ratio and then the results obtained from the same engine with a 5 to 1 compression-ratio. The results, plotted with revolutions per minute as the abscissas and the horsepower per cubic inch of piston displacement as the ordinates, give the three curves shown in Fig. 5. The reason they are brought to a horsepower cubic-inch-displacement basis is to make a direct comparison between the 295.2 and the 224.0-cu. in. engines. In the 5 to 1 compression-ratio engine, the performance falls below that which we obtained. This was surprising to us.

The comparison in percentage between the three curves is shown at the top of Fig. 5. Taking the Ansted-engine curve as 100 per cent, we find that the Nelson engine with a 5 to 1 compression-ratio, at from 900 to 1900 r.p.m., has about 2 per cent more power per cubic inch of displacement. We find that the old 4¼ to 1-ratio engine varied from 80 up to 92, and down again to 69 per cent of the Ansted-engine power, at speeds varying from 400 to 2800 r.p.m., the point at which the Nelson curve ends.

Fig. 6 is interesting as giving a general idea of how a standard engine performs, its torque curve being practically flat and running from 120 down to 105 lb-ft. at 3000 r.p.m. and down to 100 lb-ft. at 300 r.p.m. The mechanical efficiency of this engine fell off more rapidly from its low speed where it had about 98 per cent efficiency to 74 per cent at 3000 r.p.m.

Fig. 7 shows some surprising features. We used a small quadrant in which the graduations of the movement of the carbureter throttle were divided into 20 divisions on the throttle arc. The throttle is absolutely tight when closed. On the Model-M horizontal Rayfield carbureter there is a small opening that allows the engine to idle when the throttle is closed. For that reason we were able to allow the engine to idle and start out with a zero reading for the throttle-opening on all of the tests. The quadrant placed on the car while it was being run on the road to determine the throttle-opening for each car speed was graduated in 10 equal divisions. In Fig. 7 we have merely stretched them out in a straight line instead of showing them on a curve. The impressive thing is that, for the range up to 60 m.p.h. on a country road, only one-half throttle was required and, when the throttle was opened wide, the car speed only increased 10 m.p.h. The revolutions per minute of the engine correspond to the car speed and the throttle-opening. With this same dial on the dynamometer, adjusted exactly as when on the road, these same speeds were maintained with the same throttle-opening by varying the number of pounds on the dynamometer. Thus, we determined how much power was being developed at that speed and throttle-opening. At the same time, an-

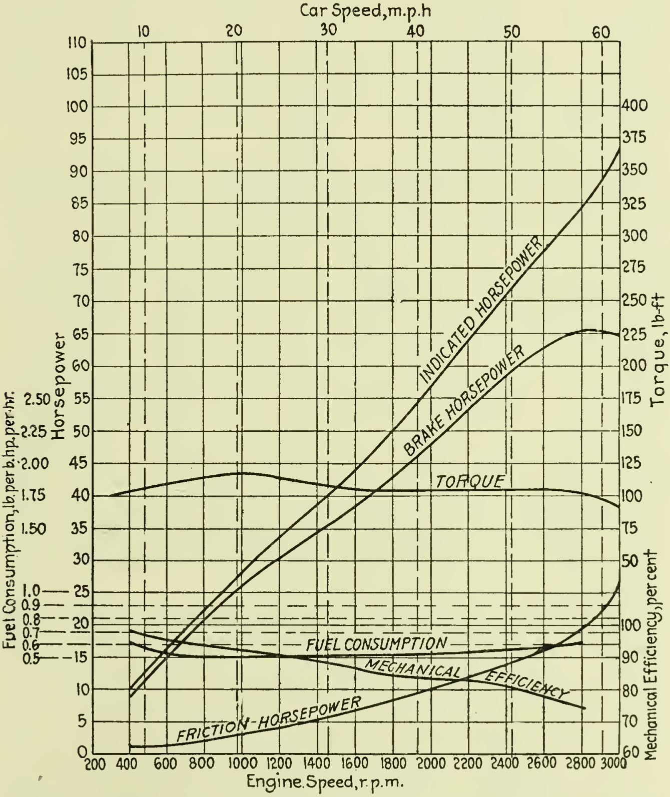

FIG. 6—PERFORMANCE CHARACTERISTICS OF THE ANSTED ENGINE

other set of tests was made that was equally significant. In trying to determine what one-quarter, one-half and three-quarter throttle-openings are, we found that nobody that had made previous tests had stated specifically how he had found what each such opening was with a butterfly throttle; so, we made four runs, taking one-quarter, one-half and three-quarters of the dynamometer load for three runs and then made a wide-open throttle run. That gave constant loads, not throttle-openings, that varied in proportion to the wide-open-throttle load. We merely ran the power curve for the maximum brake-horsepower, and then ran a series of other horsepower curves at three-quarters, one-half and one-quarter of whatever the ordinate was. We then determined the throttle-opening at those particular speeds during that run. With one-quarter load, the throttle is never opened more than one-third; with one-half load, it is open scarcely more than four-tenths; and, at three-quarters load, it is only a little more than half-open. One can almost draw a straight line through the same speed-points and throttle-openings.

After the road test, we came in, set the throttle as shown in Fig. 7, brought the revolutions per minute up to the same value and then read the pounds of torque shown by the dynamometer. From that result, we obtained the curve for road horsepower shown in Fig. 8. The abscissas are in terms of hundreds of revolutions and the miles per hour that the car travels. This is determined easily because we used a fixed gear-ratio. About 3 hp. is required at 10 m.p.h. to keep a car rolling steadily on a gravel road; about 7 hp. at 20 m.p.h.; 13 hp. at 30 m.p.h.; 20 hp. at 40 m.p.h.; 30 hp. at 50 m.p.h.; and 43 hp. at 60 m.p.h. Because we could not run the dynamometer at a speed of more than 3200 r.p.m., we did not determine what horsepower was necessary at 3400 r.p.m., which is the speed at which the combined road and wind resistances apparently balance the horsepower of the engine.

The dotted curve in Fig. 8 is the one obtained by Mr. Nelson on the Indianapolis Motor Speedway with a car weighing about 1000 lb. more than the car we used. It is particularly interesting because it shows that either our results were inaccurate or that there is greater resistance on a hard dirt road than that shown on the brick-paved Speedway. This is the first comparison made between the power required on the Speedway and on a dirt

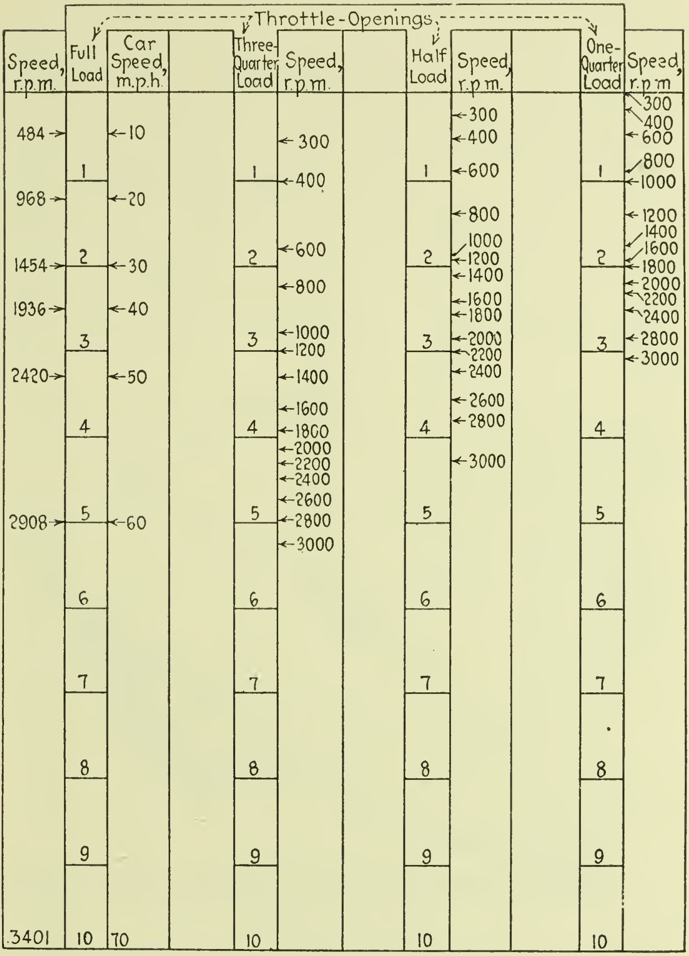

FIG. 7—DIAGRAM SHOWING THE THROTTLE-OPENINGS AT VARIOUS LOADS

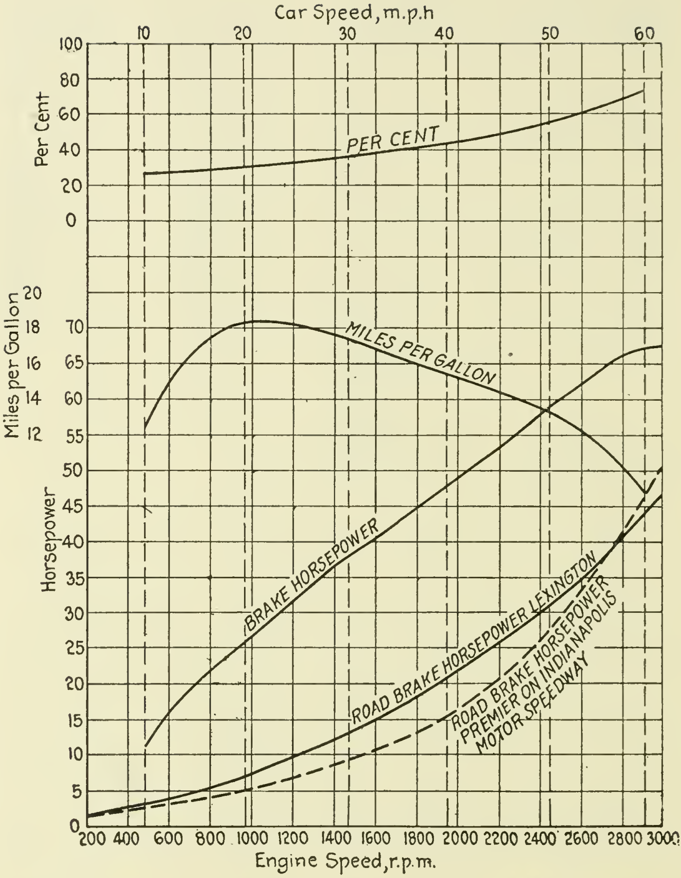

FIG. 8—ROAD CHARACTERISTICS OF A CAR EQUIPPED WITH THE ANSTED ENGINE

road. It is unfortunate that the same car was not used in both instances.

The maximum brake-horsepower curve of the Ansted engine is shown in Fig. 8. It is evident that the power available and that required to run on the road are very different. The difference in power is the amount that is available for climbing hills and for acceleration. The percentage curve at the top of Fig. 8 indicates what it is. It is the ratio between the power required on the road and the brake horsepower available. At 10 m.p.h., about 22 per cent of the power is used to drive the car, but at 60 m.p.h. about 64 per cent is required.

An interesting miles-per-gallon curve is shown in this diagram. The values were obtained from the pounds of fuel used per brake horsepower when running to obtain the road curve shown below it. From these computations, we find the values to be 18½ miles per gal. at 20 m.p.h.; about 13 miles per gal. at 10 m.p.h.; about 17 miles per gal. at 30 m.p.h.; 15 miles per gal. at 40 m.p.h.; 13 miles per gal. at 50 m.p.h.; and 8½ miles per gal. at 60 m.p.h. These values are a close check on the average runs made by this car on the road. Driving between 20 and 40 m.p.h. day in and day out, we obtained somewhere between 16 and 17 miles per gal. This is an average that would be indicated by the curve.

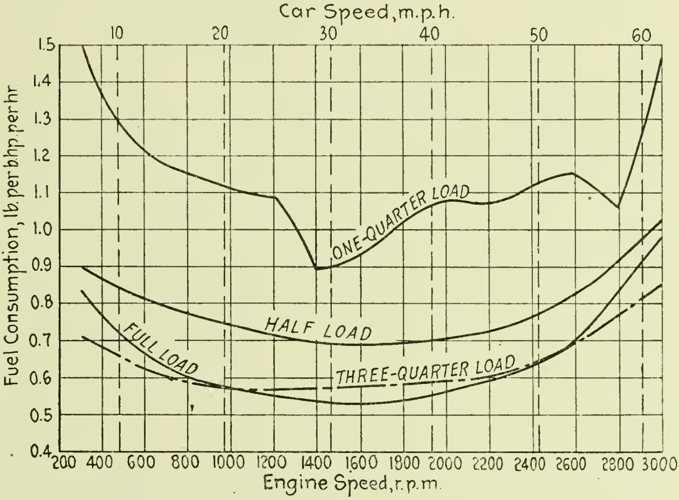

FIG. 9—FUEL-CONSUMPTION CURVES AT VARIOUS ENGINE LOADS

Some of the results obtained from the laboratory test when running the engine at full, three-quarter, one-half and one-quarter load are incorporated in Fig. 9. These curves show the pounds of fuel per brake horsepower used at various engine speeds or at equivalent car speeds. The lowest curve shows the fuel consumption obtained with full-throttle. The other three curves are significant. The one-quarter-throttle curve is of such a character that we do not wish to interpret it at this time. We followed the points precisely as we found them on that curve and the other curves fell very close to the points as they were obtained during the test. The one-quarter-load curve shows some very unusual character-

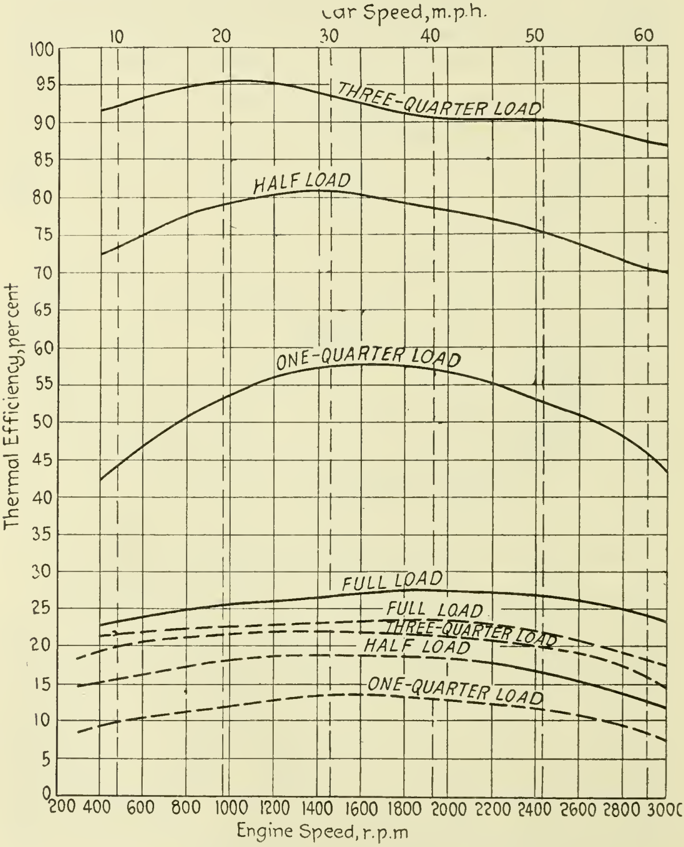

FIG. 10—COMPARISON OF BRAKE-THERMAL EFFICIENCIES AT VARIOUS LOADS

istics, indicating that something irregular happens in the carbureter and in the manifold at low-throttle.

The indicated thermal efficiencies are plotted on both the indicated-horsepower and the brake-thermal-efficiency curves in Fig. 10. The former are in full lines and the latter are dotted. They are particularly absorbing to power engineers. We do not realize how much the modern automobile-engine can do. On the indicated thermal efficiency this engine utilizes about 27½ per cent of the fuel available when valuing the fuel at 19,500 B.t.u. per lb. of gasoline. We were using 59-deg. Baumé gasoline weighing about 6¼ lb. per gal. These curves were computed on that basis. The full-load curve for the thermal efficiency on the brake horsepower runs up to 24 per cent between 1800 and 2000 r.p.m. The curves that are below it are the thermal-efficiency curves at one-quarter, one-half and three-quarter loads. The three-quarter-load thermal-efficiency is almost as good as the full-load thermal-efficiency. The one-half-load curve begins to fall off rapidly and the one-quarter-load curve drops to as low as 14 per cent at the best point; but when we remember that a large powerplant runs at only about 25-per cent thermal-efficiency with everything in its favor, a small 70-hp. gas engine does well to show equally good thermal efficiencies.

For convenience in comparing the curves, percentage curves are also shown. Assuming that the full-load efficiency is 100 per cent, the three-quarter-efficiency curve at a certain point runs from 92 to 96 per cent of the full-load efficiency, and does not drop below 90 per cent under 50 m.p.h. At one-half load it runs down considerably more, from 72 per cent at low speeds up to 81 at 30 m.p.h. and then down to 70 per cent at 60 m.p.h. At one-quarter load, at which a car operates most of the time, this curve shows that, at 10 m.p.h., the operation is at 42½ per cent of full-load efficiency. At 30 to 40 m.p.h. we may obtain as high as 57 per cent of the full-load efficiency, but it drops off rapidly as the speed increases.

Fig. 11 shows a set of curves plotted from readings taken as the experiments progressed. A tube runs into the intake-manifold and is connected to a manometer indicating the inches of mercury depression in the intake-manifold at full load with wide-open throttle. At 3000 r.p.m. there is only a 2-in. depression, and less than ½-in. depression at 300 r.p.m. The unusual characteristics are found in three-quarter, one-half and one-quarter-load in-

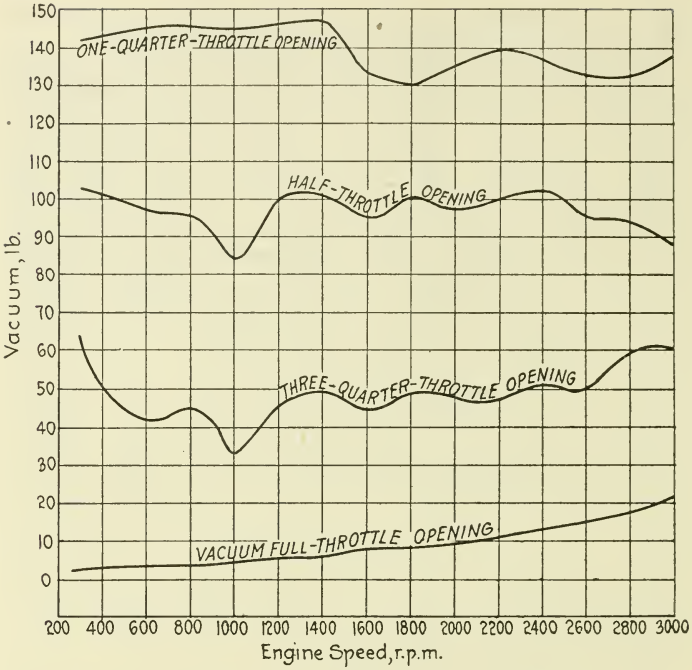

FIG. 11—INTAKE-MANIFOLD DEPRESSIONS AT VARIOUS LOADS

take-manifold pressures. The only reason that we followed the points exactly as we obtained them was that there seemed to be this general characteristic of depression in the curve at 1000 and again at 1600 r.p.m.

THE DISCUSSION

ROBERT W. CARINGTON:—What were the temperatures of the intake and exhaust-manifolds?

CHESTER S. RICKER:—We did not take their temperature readings.

G. U. SMITH:—What was the amount of compression in pounds per square inch?

MR. RICKER:—At present, it reaches 80 lb. per sq. in.

PROF. DANIEL ROESCH:—Were readings taken at different speeds so that an idea can be given of what constitutes low, medium and high speed?

MR. RICKER:—No readings were taken at any speed except that at which the starter cranks the engine; that is about 150 r.p.m.

C. H. Kirby:—What is the peculiarity shown in that depression or vacuum curve?

Frederick Purdy:—I am unable to explain the curious curves shown in Fig. 11. What we call a fractional opening of the throttle is not a fractional division of the number of degrees; it is a certain metrical function of the cycle. I believe one-quarter of that, then another quarter and finally four-quarters would give the full opening, and that would give also the uniform area. That would account also for the half-open throttle at 60 m.p.h. and all the remainder of the way, because the increase of area is not very considerable.

When I first saw those curves, I thought that the velocity effect on the exploring tubes must have been the cause of an erratic reading; for, in the readings as they were taken, with a tube extending into the manifold and with the opening at right angles to the axis of the manifold at that point, the static or the true suction-head and also the effect of the velocity or the dynamic head would be shown; that is, if one were taking the minus pressure as a suction value. But on further consideration this seems not to be tenable. As I understand it, there was no variation in the throttle-opening or in its position, and the only variation was a load on the dynamometer to change the speed; so, it seems that we could not have any such curious shapes as those to atone for. I thought also that the drop in the suction value might have been at a point where the air-valve opened, but this seems to be so pronounced and so uniform that it is a characteristic. There seem to be two drops. With this point somewhat further along and nearer the slow side, it might possibly be accounted for by the point at which the valve began to open. There would be a gradual rise in the suction value as the engine speed increased, and then a sudden drop as the valve broke its position. If that curve were laid out in terms of energy to speed and were a superimposed curve of the suction, there might be some sort of correspondence there, but the curious shape is a mystery to me.

Mr. Ricker:—Concerning air-valves, with a hot-spot "frying grid" such as is used in the manifold on this engine, as shown in the cross-section of the head in Fig. 3, the fuel is fed into that hot-spot and vaporized. It is thrown in there by the inertia due to the sudden change in the direction of the gas. The thing to be noted in connection with that type of manifold is that an almost dry gas enters the cylinders under all conditions. The only thing to watch out for is a sudden starving of the engine when the accelerator pedal is pushed down all the way in attempting to get a quick start from low speed.

The next thing that had to be considered was enough port-opening to permit the high engine-speeds that are obtainable. It was essential to get a range of from 300 to 3000 r.p.m. in a stock engine. Without an air-valve type of carbureter it seems to be impossible to obtain a speed range of from 300 to 3000 r.p.m. Because the air column is so much lighter than the fuel column issuing from the jet, the former goes into the engine and leaves the fuel behind when the throttle is kicked wide-open at low speeds. Unless something is done to dampen or slow up that air-column movement for the instant when the throttle is kicked open, one cannot obtain anything like a uniform mixture. That accounts for the good acceleration with this engine from such low speeds as 2 m.p.h. without a flat spot in the curve, and the fact that the carbureter can be adjusted for economy while still giving the fuel-efficiency curves shown, without any readjustment during the tests. I am not saying that this carbureter is the only one that will give such performance but, from my experience with this and other engines, I feel that the air-valve type of carbureter merits very serious consideration for the maximum speed-range type of engine.

ROY E. BERG:—Were gas temperatures recorded during the runs?

MR. RICKER:—No.

MR. KIRBY:—How did you make the determination of gas speeds?

MR. RICKER:—The gas speed is the number of cubic inches of displacement per minute divided by the area of the manifold section in square inches and divided by 12 to reduce it to feet per minute. The gas velocities given in Mr. Nelson's paper were figured for one-half the engine displacement for each side of his longitudinal manifold. Using those values here, we obtain about one-half the velocity Mr. Nelson reports.

A MEMBER:—Were these tests made with the radiator in place and the cooling system operating just as on a stock car?

MR. RICKER:—Not on the dynamometer test. The fan was the only part of the system not in operation. We had practically the same resistance as in the radiator.

The engine is maintained at a constant temperature by thermostat; so, irrespective of whether it was on the car or on the dynamometer, the temperature of operation was maintained constant.

A MEMBER:—Was the thermostat connected to a pressure system or to a tank?

MR. RICKER:—To a tank.

A MEMBER:—Were those manifold-pressure curves taken at a fixed throttle-opening or at a fixed torque?

MR. RICKER:—The torque varied; the throttle-opening was constant.

DENT PARRETT:—Without offering any criticism of the carbureter, it occurs to me that possibly there is a point in this changing load where a leaner mixture is fed to the engine. For that reason, to pass this horsepower equivalent to a basis of 1000 r.p.m., the manifold pressure would be reduced; in other words, the suction would increase so as to provide a sufficient volume of fuel. It simply means that the carbureter was taking in a little more air in proportion to the fuel, to overcome that point.

MR. SMITH:—What were the ignition curves corresponding to the throttle-openings?

MR. RICKER:—We recorded the data but they have not yet been charted. They varied, however, and we maintained them the same on both the road tests and the dynamometer tests, the spark being advanced the maximum amount permissible without causing "pinging" in the cylinders during the road tests.

JOHN W. STACK:—How were those indicated-horsepower curves plotted? Are they the result of actual observations?

MR. RICKER:—No, the indicated horsepower was taken as the sum of the brake horsepower and the friction horsepower on two different engines, but taken at the same engine speeds.

F. G. SHOEMAKER:—In connection with the curves in Fig. 11, I have observed two things in dynamometer testing. For instance, in taking vacuum readings at partial throttle-openings with and without gasoline, that is, setting to a fixed throttle-opening and turning the mixture on, the pressure in the manifold changes considerably when gasoline is used and when there is no gasoline. For that reason, changes in mixture proportions might account for some of the changes in the curves. Another thing is that we set up resonant periods in intake-manifolds at certain speeds. Might not this be accounted for by resonance in the intake pipe? I have noticed that this has a strong effect in the exhaust-manifold. We had a long 5-in. pipe on the exhaust in our dynamometer equipment that extended about 10 ft. and the horsepower curve dropped off about 2 hp.

MR. PARRETT:—Would not the fact that the dip in the curve occurs at 1000 r.p.m. in both cases be an argument in favor of the idea that this resonance accounts for the effect?

GEORGE E. MARTIN:—Several late designs of engine have shown greater speed and horsepower than earlier engines of the same type. What is the limiting factor of the engine speed and horsepower that can be developed in an engine of a given size? Is it a matter of the amount of fuel that can be put into the engine and burned, or the method of introducing the fuel, or a mechanical characteristic of the engine?

MR. RICKER:—That can be answered by relating some experiences we have had with this engine. Experimental work shows that an increase in valve size or lift gives no further increase in the power. We are trying to determine how much a greater manifold size will do toward increasing the power. The limitations seem to be almost entirely with the breathing apparatus. If the breathing apparatus were large enough, we might get all the power desired from the cylinders by allowing the engine to turn faster and faster. We may encounter balance and inertia effects that would require the use of some other type of piston, probably a lighter reciprocating part if the speeds were increased beyond this, but we have every reason to believe that if more power is desired there is a possibility of obtaining it by increasing the size of the breathing apparatus without otherwise changing the inherent design of the engine.

MR. MARTIN:—Horsepower efficiency is based on the British thermal units in the fuel in steam-engine practice. If a point could be reached where 100 per cent of the British thermal units in the fuel were utilized, would not that be the limiting factor in what might be developed in a gas engine?

MR. RICKER:—Judging from Diesel-engine practice, we know that the theoretically perfect engine operating on the constant-pressure cycle, assuming air-standard efficiency, develops mechanical power that represents only 57 per cent of the available British thermal units in the fuel. Therefore, the Diesel engine, having an efficiency of say 40 per cent or better, is really utilizing 80 per cent or more of the available heat units in the fuel. On that basis, if 27-per cent efficiency is being obtained as shown by the indicated-horsepower curve, we are really obtaining about 55 per cent of the available power in the fuel, and this is far from being an ideal engine performance.

MR. PURDY:—The very fact that the valley of the curve, not the peak, occurs at a fixed engine speed rather than at a suction value, indicates that resonance is an effect that is responsive to speed and not to suction. The carbureter cannot take cognizance of the engine speed; it only takes cognizance of the suction value. So, if it were due to something within the carbureter or to some change that takes place inside the carbureter, that characteristic drop would not be at the same engine speed.

MR. SHOEMAKER:—Perhaps that resonance is in the air-valve of the carbureter, and we can attribute the effect to the carbureter.

MR. PURDY:—Undoubtedly, but that resonance effect would be responsive to suction rather than to engine speed because, with some particular carbureter, such as the one used in this instance, it is so well damped that resonance in the mechanical part of the carbureter is rather out of the question. Resonance in the moving column of air would be, of course, the same in one type of carbureter as in another, assuming that it is not modified by the resonance or reaction of the mechanical moving parts.

MR. KIRBY:—Were the manometer readings steady at the time?

MR. RICKER:—They were practically steady.

MR. KIRBY:—Were they uniform throughout the range?

MR. RICKER:—Yes.

PROFESSOR ROESCH:—That would cure the resonance effect. Would not the same effect occur at a speed of 2000 r.p.m.?

MR. RICKER:—There is a slight effect at 1600 r.p.m.

MR. PARRETT:—It may be due to the effect of the pulsations in the suction of the engine combined with the action of the air-valve. At some certain speed the valve would fluctuate more and, if the valve were closed part of the time, that would generate suction and cause this drop in the curve.

PROFESSOR ROESCH:—Why is that manifold constructed with square ends, as shown?

MR. RICKER:—It has a hole straight through with the ends closed by core plugs. It is done as a matter of manufacturing convenience to support a core at each end.

PROFESSOR ROESCH:—Is that the best design and was it determined experimentally?

MR. RICKER:—It is a matter of manufacturing convenience, but I believe it has considerable merit.

PROFESSOR ROESCH:—In the application of the friction-horsepower curve in Fig. 6 to the brake-horsepower curve in Fig. 4, was that friction-horsepower curve taken on a stiffer engine?

MR. RICKER:—Yes, it was taken on a very much stiffer engine.

PROFESSOR ROESCH:—Then the indicated-horsepower curve would be too high in Fig. 4.

MR. RICKER:—Yes.

PROFESSOR ROESCH:—In connection with measuring the throttle-openings shown in Fig. 7, were the 10 equal divisions measured on the steering-post?

MR. RICKER:—No. The measurements were made directly on an indicator attached to the throttle-valve arm, and the sheet was accurately fixed on the carbureter itself. A sheet-metal support was made for it, so there was no question as to its positioning with respect to the center of the throttle-arm. The radius of the points was about 5½ in.; so there was a fairly small movement of the throttle-valve itself and a very large indication on the dial. The range is slightly more than 90 deg., without any lost motion.

PROFESSOR ROESCH:—In connection with those small throttle-openings required to drive a car at from 400 to 800 r.p.m. of the engine, the observation of the intake-manifold suction might be checked, because it extends over a rather wide range and is measured more easily than are smaller dial ranges.

ALUMINUM PISTONS[1]

By Ferdinand Jehle[2] and Frank Jardine[2]

The lightness and high thermal conductivity of aluminum pistons are conceded and the paper deals principally with their thermal properties, inclusive of the actual operating temperature of the pistons, the temperature distributions in the piston and the effects of the cooling-water temperature and the piston material on the piston temperature. The apparatus is illustrated and described, and charts are presented and commented upon in connection with a discussion of the results obtained.

Theories affecting piston design are presented and discussed, reference being made to diagrams relating to design procedure. The work is supplementary to that done in 1921 by the authors, which they presented in a similar paper to which they refer.

Aluminum possesses two inherent advantages over all other metals that are used for piston materials, lightness

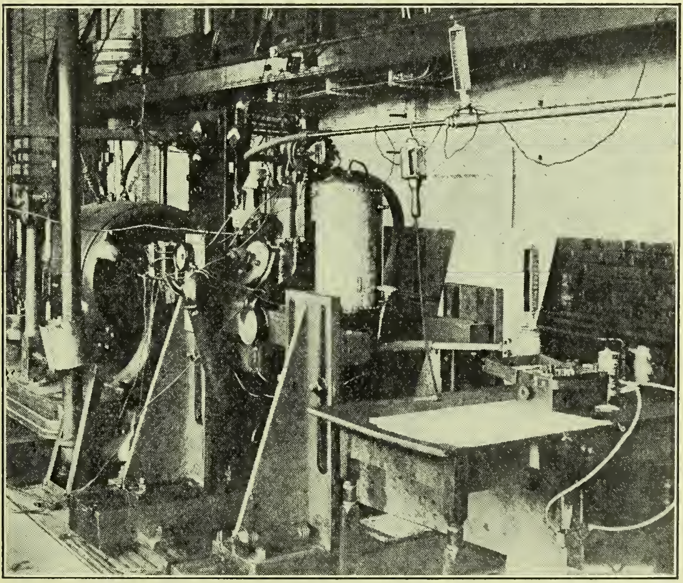

Fig. 1—Apparatus Employed To Measure Piston Temperatures

and high thermal conductivity. Engineers and the public in general realize the value of lightness in a piston, as is proved by the various thin gray-iron pistons now being manufactured, the weight of which approaches that of the aluminum pistons. The advantages gained by making the piston a good heat-conductor are not appreciated so thoroughly as the desirability of lightness, which again is demonstrated clearly by the numerous light iron pistons. This paper deals principally with the thermal properties of the piston, such as the actual operating temperature of the piston, the temperature distribution in the piston, the effect of the cooling-water temperature on the piston temperature and the effect of the piston material on the piston temperature. We hope that it will be received as an additional chapter in the work of piston development.

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[1] Detroit Section paper.

[2] M.S.A.E.—Engineer, Aluminum Manufactures, Inc., Cleveland.

In studying piston temperatures, the same set-up was used that is described in our previous paper on Aluminum Pistons.[3] Fig. 1 shows the apparatus. It consisted

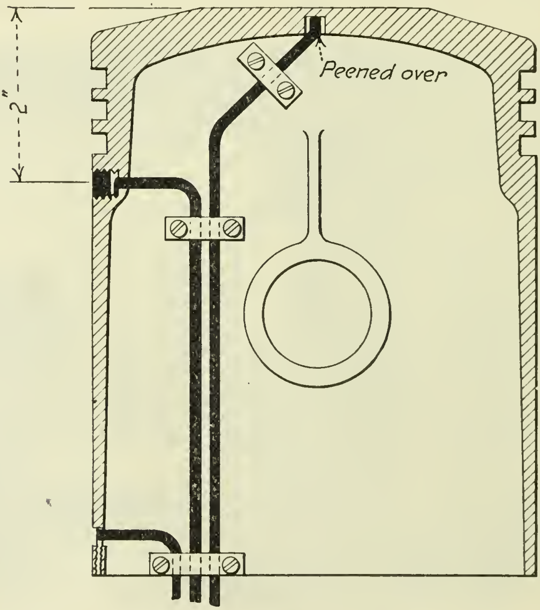

Fig. 2—Method of Fastening Thermocouples in Place in the Piston

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[3] See Transactions, vol. 16, part 1, p. 656.

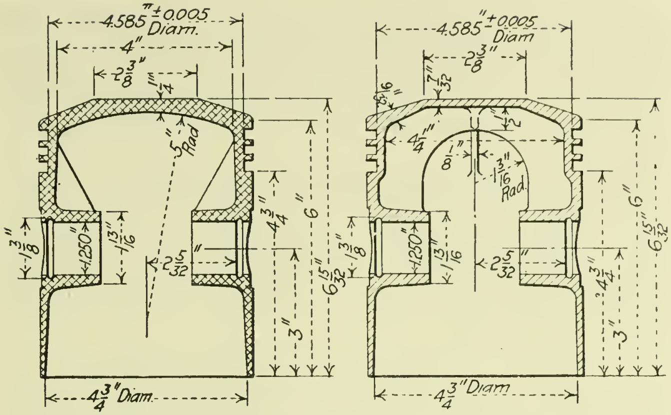

Fig. 3—Details of the Two Pistons Tested

The Piston at the Left Was Made of Aluminum While Cast Iron Was Used for That at the Right

principally of a single-cylinder Liberty engine, a Sprague dynamometer and temperature-measuring equipment. The details of the temperature-measuring apparatus were illustrated in the previous paper and are not repeated here. The center head thermocouple was fastened by a slightly different method from that used in the previous work. In those experiments the head thermocouples were held in place either with a screw plug or a welded plug inserted from the outside of the piston. In the present work the head thermocouple was inserted by drilling a small hole only slightly larger than the thermocouple wires into the head, from the inside, for a short distance. The welded end of the couple was inserted in this hole and the surrounding metal was then peened over. Fig. 2 shows the method of fastening. The thermocouple had to be clamped securely to the piston, since the peened joint possessed no mechanical strength. By using this fastener for head thermocouples rather than the previous ones, there was absolutely no chance for hot gases to leak past the thermocouples and thus give a temperature reading higher than the true temperature of the piston-head. A comparison of the present with the former results indicates that the new method of thermocouple fastening was an unnecessary precaution. The thermocouples in the piston skirt were held in exactly the same manner as in the previous work, since there is no possibility of gas leaks at these points. During all of the tests reported in the present paper the engine speed was held at 800 r. p. m., with wide-open throttle and maximum spark-advance. The power developed was between 12 and 13 b.hp. Care was taken to keep the fuel-consumption constant throughout the tests. Only readings obtained after engine conditions were constant were used in the computation of the results.

The view at the left of Fig. 3 is a detailed drawing of the aluminum piston used. It was 5 in. in diameter and had a head ¼ in. thick, with ample metal behind the rings. Without rings or pin it weighed 3 lb. Attention is called to the fact that this was not a regulation Liberty-engine piston.

EFFECT OF COOLING-WATER TEMPERATURE

A rather thorough study was made of the effect of the cooling-water temperature on the piston temperature. In the previous paper enough work was done on this subject to lead one to believe that the cooling-water temperature had an important effect upon the piston temperature. In the earlier work, piston-temperature readings were taken at only two temperatures of ingoing cooling-water; namely, 126 and 48 deg. fahr. In the present work, the highest temperature of ingoing cooling-water was approximately 160 deg. fahr., and measurements of piston temperatures were made at intervals of about 20 deg. until an ingoing-cooling-water temperature of 70 deg. fahr. was reached. The results obtained absolutely checked-up the indications of the previous work. The curves on Fig. 4 represent the temperatures of several points on the piston, plotted against the ingoing-cooling-water temperatures. The insert in the figure shows the points at which the temperature was measured. The temperatures of three points were determined; namely, in the center of the head, in the skirt immediately below the last ring and in the lower extremity of the skirt. The skirt temperatures were taken on the intake side of the piston.

The temperature of the head is affected most by a change in the cooling-water temperature; that is, a given change in the cooling-water temperature brings about a change in the head temperature of a greater number of degrees than the same change in ingoing-cooling-water temperature produces in the piston skirt. However, the rate of change is of the same general magnitude. It will be noticed that the curves flatten out at the higher cooling-water temperatures, which indicates that the piston temperatures under these conditions are not so susceptible to a change in the cooling-water temperature as they are at lower cooling-water temperatures. This would lead one to believe that the piston-head temperature in an air-cooled engine might not be so very much higher than in a water-cooled engine utilizing high cooling-water temperatures.

Comparing the different curves in Fig. 4, it is brought out clearly that there is an enormous drop between thermocouple No. 1, at the center of the piston-head, and thermocouple No. 3, immediately below the last ring. The drop from thermocouple No. 3 to thermocouple No. 4, that is, over the skirt, is not great. Unfortunately, thermocouple No. 2, immediately above the rings, was not operative. However, in the previous work the drop across the head was never 50 per cent of the drop across the rings and, since all of the measurements taken coincide so closely with the previous ones, it can be assumed safely that the greatest temperature-drop is across the rings.

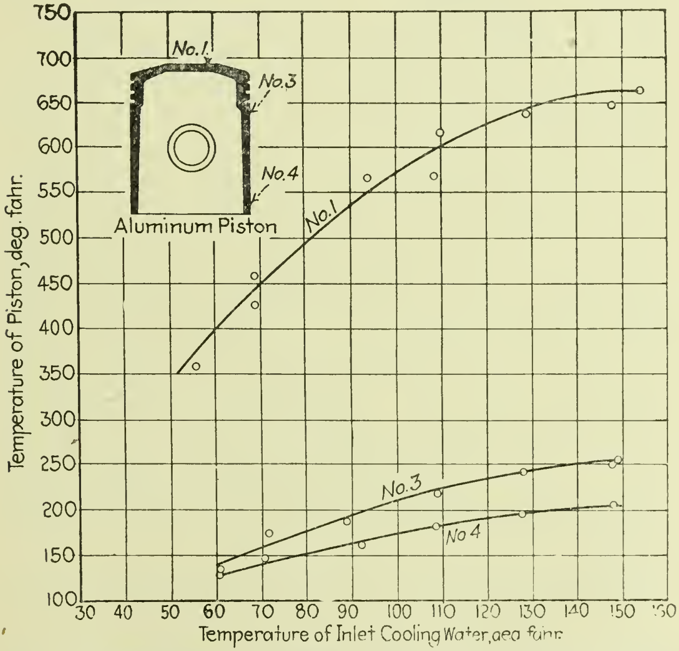

FIG. 4—TEMPERATURES AT SEVERAL POINTS ON THE PISTON PLOTTED AGAINST THE COOLING-WATER TEMPERATURE

PISTON MATERIAL CONDUCTIVITY EFFECT

In our previous paper a short series of tests was reported that showed the difference in the piston temperature to be due to a difference in metal thickness; that is, in the conductivity of the piston. Reference is made to some measurements of piston temperatures in a Diesel engine that were reported.[4] The pistons were made of cast iron and measured 400 mm. (15.75 in.) in diameter. Two pistons were tested, the difference between them being in head thickness. One had a head thickness of 52 mm. (2.05 in.); the other had a thickness of 61 mm. (2.40 in.) Under the same conditions of load and the like, the thinner piston showed a head temperature of 443 deg. cent. (829 deg. fahr.), while the thicker one showed a head temperature of only 402 deg. cent. (756 deg. fahr.). Unmistakably, the thicker the piston-head and the walls are, the lower the head temperature will be. This can be taken to mean that the higher the conductivity of the piston is, the lower the piston-head temperature will be, regardless of whether this increase of conductivity is brought about by thickening the section or by using a material possessing better thermal properties.

To prove definitely that cast-iron pistons operate at a higher head temperature than aluminum ones, we made some temperature measurements of a cast-iron piston running in the single-cylinder Liberty engine. The piston was operated under the same conditions as the aluminum piston while the results shown in Fig. 4 were being obtained. The drawing at the right of Fig. 3 gives details of the cast-iron piston, which is of somewhat lighter construction than the aluminum piston and has two ribs. These pistons represent about the usual relation between the cast-iron piston and the aluminum one that is used to replace it. The former weighed 5.9 lb. and the aluminum piston only 3 lb. Temperature measurements of this piston were made at different ingoing-cooling-water temperatures similar to those made on the aluminum piston. Fig. 5 is a chart that shows the difference between the cast-iron-piston temperatures and the aluminum-piston temperatures; they are just about what was expected.

The head temperature of the cast-iron piston is more than 200 deg. higher than that of the aluminum piston operating under the same conditions. For example, comparing the temperature of the two pistons when the ingoing-cooling-water temperature is 160 deg. fahr., we find that the head of the aluminum piston reaches a temperature of 660 deg. fahr. under these conditions, while the cast-iron piston-head registers 880 deg. fahr. When the ingoing-cooling-water temperature is reduced to 75 deg. fahr., we find the temperature of the aluminum piston-head is 420 deg. fahr., while that of the cast-iron piston reaches 660 deg. fahr. In other words, the cast-iron piston-head was as hot when the ingoing-cooling-water temperature was 75 deg. fahr. as the aluminum piston was at an ingoing-cooling-water temperature of 160 deg. fahr. A high piston-head temperature brings about certain well-known bad results such as carbonization above and below the piston head and preignition, which may necessitate lowering the compression. The point might be raised that, since the cast-iron piston is as hot with the cooling water at 75 deg. fahr. as the alu-

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[4] See *Zeitschrift des Vereines Deutscher Ingenieure,* Aug. 27, 1921, p. 923.

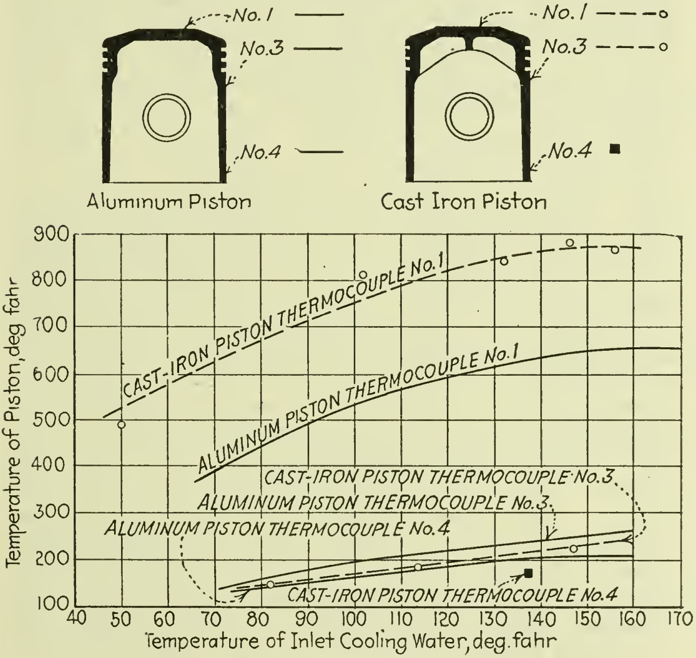

Fig. 5—Comparison of Temperatures of the Cast-Iron and Aluminum Pistons Plotted Against the Cooling-Water Temperature

minum piston is when the cooling water is at 160 deg. fahr., an engine equipped with cast-iron pistons would operate better on a cold day than one equipped with aluminum pistons. Experience indicates that this is not the case. As A. L. Nelson points out in his paper on the Fuel Problem in Relation to Engineering Viewpoint,[5] a hot cast-iron piston-head does not assist vaporization more than an aluminum piston-head because the heat content of the two is about equal and the heat flow from the latter is much greater.

The temperature distribution in the cast-iron piston is somewhat similar to that which is obtained in the aluminum piston, except that it is exaggerated. The drop across the rings in the cast-iron piston is much greater; in fact, it is greater by something over 200 deg. fahr. than it is in the aluminum piston. The actual skirt-temperature is somewhat lower in the cast-iron piston. This can be accounted for by the fact that the thermal conductivity of cast-iron is much less than that of aluminum. Only one temperature-measurement was made on thermocouple No. 4; that is, at the bottom of the skirt. This great difference of temperature between the heads of cast-iron and aluminum pistons suggests that possibly the field for aluminum pistons is not limited to the high-speed passenger-car engine. The aviation engine is not generally considered a high-speed engine, yet aluminum pistons are used in the great majority of such engines. It can be stated that in aviation-engine practice the aluminum piston has not been used primarily for its lightness, but because of its excellent thermal properties. The motor-truck engine is not a high-speed engine; but it is a heavy-duty engine; and so is the tractor engine. As a rule such engines are equipped with cast-iron pistons of fairly heavy section. The heating conditions in a truck and in a tractor engine are far more difficult to handle than in the passenger-car engine which runs at a higher speed. Much could be gained in the performance of slow-speed heavy-duty engines by the use of aluminum pistons. The cooler piston-head would permit higher compressions without a doubt, and would reduce carbon deposit and all its attendant evils. Incidentally, it would eliminate much vibration and many evils that accompany vibration, such as the loosening of joints. A reduction of bearing pressures would be assured also, and this is very desirable.

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[5] See TRANSACTIONS, vol. 16, part 1, p. 325.

THEORIES AFFECTING PISTON DESIGN

The general theory upon which the separation of the piston-head from the skirt is based is given in detail in our previous paper. It is that the expansion of a piston skirt is due to two things; (*a*) the thermal expansion due to the temperature of the skirt and (*b*) the mechanical distortion of the skirt caused by the thermal expansion of the head.

To minimize the effect of the expansion of the head upon the skirt, circumferential slots separate the thrust faces from the head. This permits the head to expand free from the separated portions of the skirt. The sides of the skirt attached to the head are relieved to make the expansion in that direction harmless. The slotting of pistons in one way or another is not new, although the application to the aluminum piston, in conjunction with other things, is of more recent date. It has been found that a piston equipped with circumferential slots, as shown in Fig. 6, has a harder bearing along the edge of the bearing surface than along the center of the thrust face. This difficulty can be remedied.

Fig. 7 illustrates a piston that has circumferential slots

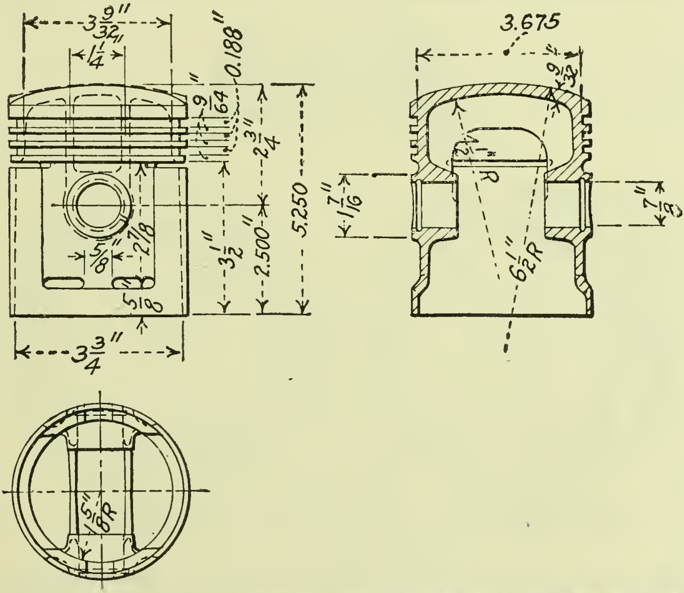

FIG. 6—IN THIS PISTON THE MECHANICAL EXPANSION DUE TO THE HEAD IS TAKEN CARE OF BY RELIEVING THE SIDES

separating the head from the skirt for a certain portion of its circumference. The view on the left-hand side is a section on *A A* of such a piston, which was machined to a true circle when cold. When the piston is heated during its operation in an engine, the effect of head expansion along the center-line *B B* of the slotted portion of the skirt is almost negligible; in fact, it may be negative. The expansion of that portion of the skirt connected to the head is exactly the same as in a piston without any slots. Therefore, while the piston is in operation, the portion of the skirt that is slotted away from the head is no longer a true circle having its center at the center of the piston. The radius *O C* to the edge of the slot has increased in length more than the radius *O B* to the center of the slotted area. The two thrust faces are, therefore, somewhat oval in shape and the piston will show a heavy bearing or score-marks along the edges at *C*. The dotted lines show the shape of the piston when it is in operation.

It is easy to give the piston skirt such a shape when cold that it will have the proper shape when heated during operation in an engine. In Fig. 7 it has been assumed that the proper shape of the heated piston is a circle concentric with the center of the piston. The right-hand

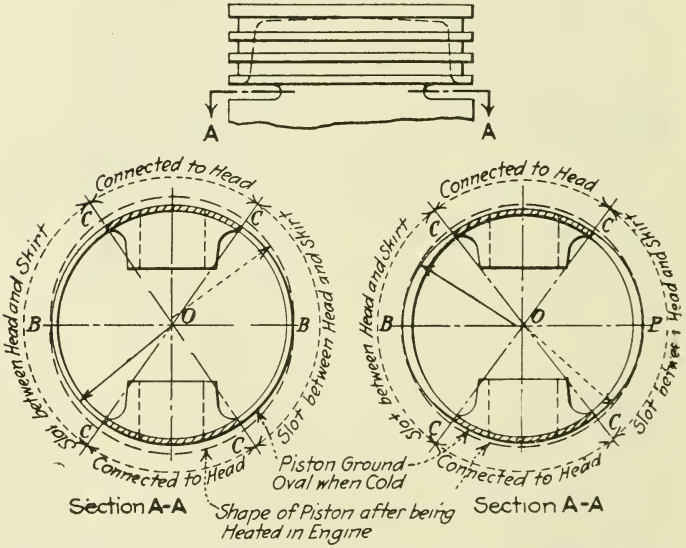
A
A
Connected to Head
Slot between Head and Skirt
C
B
O
Connected to Head
Section A-A
Connected to Head
Slot between Head and Skirt
C
B
O
P
Piston Ground Oval when Cold
Shape of Piston after being Heated in Engine
Section A-A

FIG. 7—TWO FORMS OF PISTON IN WHICH CIRCUMFERENTIAL SLOTS SEPARATE THE HEAD FROM THE SKIRT FOR A PORTION OF THE CIRCUMFERENCE

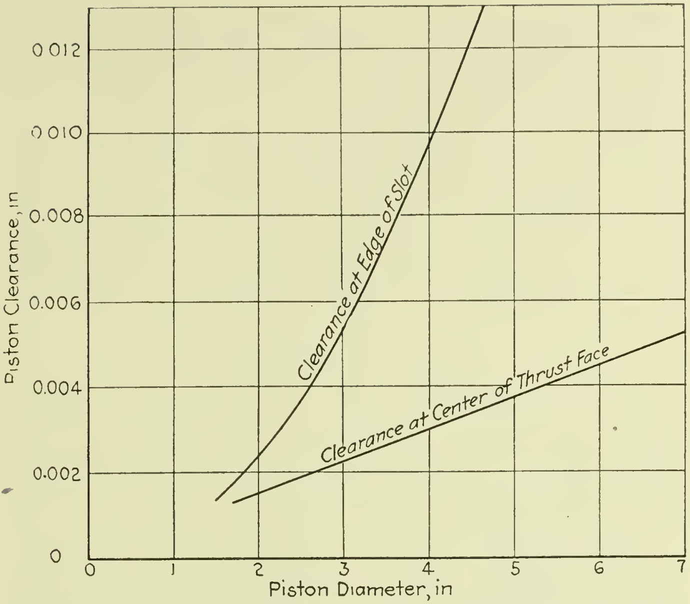

FIG. 8—CURVES GIVING THE PROPER CLEARANCE FOR SLOTTED-OVAL PISTONS

view of the section on *AA* in Fig. 7 illustrates a piston machined oval for the entire length of the skirt to such a degree that it will expand into a true circle in operation. Again the dotted line shows the shape of the piston skirt in operation.

There are, of course, several forces at work controlling the expansion of a piston skirt and, even if the temperatures of the skirt and of the cylinder were known, it would be a very complicated problem to calculate the exact amount of clearance for the different portions of the skirt. These clearances must be determined experimentally. It is reasonable to expect that the point *C* in Fig. 7 on the bearing surface of the piston at the edge of the circumferential slots should have the same clearance as the ordinary trunk-piston, and this actually has worked out in practice. A very much smaller clearance can be used on the center of the bearing surface. The amount necessary has also been worked out experimentally. Fig. 8 shows two curves; the upper curve gives the clearance to be used at the edge of the slot as at *C* in Fig. 7, and the lower one gives the clearance for the center of the thrust face as at *B* in Fig. 7. While the curves in Fig. 8 are based upon experiments, the exact clearances necessary depend upon the type of engine and may, of course, vary somewhat.

It must be borne in mind that the edge of the bearing surface may not coincide with the edge of the slot as in Fig. 6. In this case the bearing surface is given the proper shape by drawing an arc between the point *B* and the imaginary point *C* in Fig. 7, proper clearances, as in Fig. 8, having been allowed at both points. The machining of such pistons need present no difficulties, since they can be ground easily by using a master cam.

THE DISCUSSION

A. A. BULL:—Perhaps the fundamental fact regarding slotted pistons is that the piston should be of such construction that it will yield instead of sticking, which is a direct indication that it must expand. It is most undesirable that a piston should stick, but we must recognize also that, before it sticks, the pressure existing between the piston and the cylinder-wall must necessarily induce excessive friction. This is indicated best by installing some of the so-called non-expanding pistons in an engine running at say 2400 r.p.m. with the throttle set, noting how rapidly the engine speed begins to lag and then calculating the energy that is absorbed or wasted by the friction that is set up.

In the piston design illustrated in Fig. 7 that has two circumferential slots below the lower facing, the purpose was to create a differential expansion in the piston to the extent that at least one diameter would remain constant. That truly can be called a constant-clearance piston. The service in some 5000 cars, extending over a period of 3 years, has indicated the absolute success of that type of piston and the fact that it can be fitted satisfactorily to a clearance as low as 0.001 in. in a cylinder 3½ in. in diameter.

Our methods of measuring temperatures in aluminum and cast-iron have been somewhat different, perhaps, from those that have been illustrated and explained. Our method gives figures that agree closely with those given by Mr. Jardine. We first attempted to find what the average temperature of the piston-head was under actual operating conditions. This was accomplished by inserting a wire of a known coefficient of expansion through a piston-head. The wire was of a definite length when installed, and just long enough so that it would extend and rub on the cylinder-walls and become short to an extent such that measurement made upon it would indicate the average temperature. Simulating those conditions in a stationary set-up, with a flame-temperature on the head to give the same average head-temperature, thermocouples were installed in various parts of the piston to obtain the so-called heat-dissipation curve, and to ascertain also the effects of the cylinder temperature on the temperature of the various parts of the piston.

Concerning piston-rings, it was a common belief not very long ago that aluminum pistons were notorious oil-pumpers just because they were aluminum pistons; but, to consider the essentials, it is believed that so long as we have a material in the piston that has a higher coefficient of expansion, we necessarily must create between the ring and the piston groove a clearance under operating conditions which is greater than that which is obtained when the ring and the piston are fitted cold. That. in itself, is unquestionably the sole reason for any difference between cast-iron and aluminum pistons, so far as oil-pumping propensities are concerned. That brings us to the desirability of having a ring that will at least seal the piston-ring against one side of the groove so that it can adapt itself. When that is done, the question of lubrication and groove wear will have been eliminated.

Another factor that has not been given particular attention in this discussion of pistons is in regard to what the cylinder shape is under actual operating conditions. Fortunately, we have been able to determine what it is to some extent; it depends largely upon construction, and it has been demonstrated clearly that the problem of pistons and piston-rings is simplified greatly with a cylinder constructed so as at least to maintain its shape under operating conditions.

Another common belief is that the effect of vacuum in the cylinders is to increase the oil-pumping conditions. In observations of oil-pumping and of determining the best rings to use by running the engine without any cylinder-head and therefore having no vacuum to contend with, we find that the amount of oil that can be pumped up by the piston under those conditions is infinitely greater than it could ever be under actual operating conditions. Have we not been too prone to believe that we must depend entirely on pistons and rings to effect a cure for oil-pumping? Have we given full consideration to the conditions that surround the bearings or the amount of oil in the cylinder? I think not.

A. E. DAMON:—We find that the circumferential slot around the head of the slotted piston is not there to check or retard the flow of heat down from the head of the piston to the skirt alone; it acts as a scavenger as well as a resistance to the heat flowing down to the skirt. We find also that, by adding a series of fins to the inside of the piston head which carries the boss, they conduct the heat very readily to the side-walls and the skirt, leaving the head, in the case of some of the air-cooled engines, practically cool; at least, so cool that we have been able to increase the compression and eliminate all spark-knock and preignition, this being indicative of no heat worth mentioning.

In regard to the slots being used to relieve the expansive effort of the piston against the side-walls of the cylinder, we are granting that the slot does that to a certain degree, no doubt; but, in the new type of E. C. Long piston of which I speak, the circumference of the skirt is actually smaller under heated conditions than it is when the piston is cold, practically relieving any tension whatever or any wall-pressure on the skirt or on the piston, other than enough to cause the cylinder-wall to act as a guide to the piston. There practically is no friction on the side-wall of the cylinder due to the piston skirt other than what we all know is a rolling friction of lubricating oil, which is the lightest friction it is possible to have.

F. JEHLE:—It is true that there is a difference in the side-clearance between the ring and the groove, when the piston is cold and when it is hot. It is true also that this difference is greater with the aluminum piston than with the cast-iron piston. Therefore, if the width of the ring were reduced one-half, we would have a condition that more nearly approaches the condition with the cast-iron piston. That is why I believe a narrow ring is a good thing.

DANIEL ROESCH:—Why limit the one-half-width ring to the aluminum piston?

MR. JEHLE:—I think a narrow ring in a cast-iron piston would be better than a wide one; this, however, for reasons other than to reduce the difference between cold and hot side-clearance.

Concerning heat throttling, while the slot separating the head from the skirt may throttle some heat, that is not the reason the slot is put there. Judging from the temperature measurements that were made on the piston without slots, there is not any great amount of heat that could be throttled off from the skirt; that is, most of it is thrown out by the ring section. The slot is put in primarily to give the head freedom to expand without distorting the skirt. If the small amount of heat throttling causes any difficulties, it is very easy to add a small amount of metal on the side where there is no slot, and thus compensate for the increase of thermal resistance caused by the slot.

I have never found any great difference between piston operation as regards heat, whether the piston had ribs and fins or not. Prof. F. C. Lea, who did much experimental work on pistons in England during the war, found that a multiplicity of ribs in the head of a piston merely adds to the difficulty of manufacturing it. He obtained results with numerous ribs that were no better than those obtained with some reasonable number, such as one or two ribs, according to his paper on Aluminum Alloys for Aeroplane Engines, read before the Royal Aeronautical Society.

T. J. LÍTLE, JR.:—I should like to ask how it is determined whether a so-called pressure-proof ring is better than a plain ring. I take it that this refers to the use of these special rings, in many cases to replace the badly worn plain ring, and in such a case undoubtedly such a ring would be found to give improved results. I have made a very close study of various types of piston-ring, and am not very enthusiastic over many of the so-called pressure-proof rings. Many of them are constructed with tapering wedge-shaped sections that are very small in dimensions and extremely delicate and difficult to install. I have found that many such rings break during tests and have also had considerable breakage reported from service-stations.

I have noted other types of ring with three sections that were supposed not only to expand diametrically, but also to increase their total section to fill the ring-groove to compensate for wear. There may be rings of this type that function properly, but some of them at least fail to do so, and show very uneven bearing surface against the cylinder-wall.

The point I wish to make is that it is possible to produce plain one-piece rings with very smooth bearing surface, preferably ground, carefully fitting them in the piston grooves. I prefer to use narrow rings on account of their lighter weight and their lessened tendency to hammer the piston groove and widen it. It is rather difficult to make an accurate determination of the relative values of these different types of ring even in the laboratory, unless rather elaborate preparations are made beforehand. One method that I have used very successfully consists of the following equipment:

A 100-gal. tank, filled with water, is placed alongside the engine being tested. A tube running from the top of the tank communicates with the crankcase. The breather of the crankcase is closed tightly, the engine is started and the water is allowed to run out of the tank just fast enough to indicate the absence of any pressure in the crankcase. In other words, all of the gas that flows by the piston and enters the crankcase is displaced into the tank as rapidly as it is formed, by allowing the water to run out of the tank as explained. By noting the time required to fill the tank with vapor and running similar tests with different types of piston-ring, a very good idea is obtained as to the relative efficiency. I have used this equipment to study piston design, as well as to compare different types of lubricating oil, and, while my experience has related mostly to the development of eight-cylinder engines, I believe that the above testing procedure would be just as useful in testing any type of engine.

In connection with the piston-rings used with the aluminum-alloy pistons, I have obtained the best results with narrow rings of deep section that fit the grooves in the piston closely and have 0.0005 to 0.0010-in. clearance. I prefer the narrow ring because there is a decided tendency in such a piston for a wide or heavy ring to peen and widen the groove at each end of the piston stroke. This action is more pronounced with loosely fitted than with tightly fitted rings. One manufacturer has gone to the extreme in this direction, and is using rings 3/32 in. wide.

One reason that excessive ring-wear is noted in certain engines is the rough cylinder-walls produced in some of the heavy-production shops. We believe that a cylinder surface should not only be ground with great accuracy, but be finished to absolute smoothness. This we accomplish by using very fine finishing-wheels with wide faces, and taking plenty of time for the operation. Each cylinder-bore is ground five times. The final operations are little more than polishing operations. It requires nearly 1 hr. to finish a cylinder block in this manner, but the results are well worth while. I repeat, therefore, that one of the reasons that piston-rings are short-lived is that they are used as laps to smooth-up the rough bores of some of our modern automobile engines.

C. R. MANES:—There is nothing wrong with the one-piece ring so long as it lasts. With a perfect groove and perfectly fitted rings no claims of efficiency for the multiple-piece ring over the plain ring can very well be established. The trouble is that the accuracy required to produce perfect grooves and fit rings perfectly to them is very difficult to obtain on any modern production job. Replying to Mr. Lítle's query as to how to determine that a so-called pressure-proof ring is better than a plain ring, I will say that if a man comes to me with an engine using from 5 to 10 times the oil it ought to use and I cure that engine permanently within a short time and at small expense by using pressure-proof rings, I surely have produced good results. If I can reduce his oil consumption one-half, one-third, or even one-twentieth, I have got results. We do not always get perfect results, because there are many types of engine and different conditions to be met, but in 95 per cent of the cases we know we get more than sufficient improvement in the operation of the engine to justify the expense to the owner. We do this with worn ring-grooves after the plain ring has ceased to function properly and the cylinder-walls are in many instances badly worn. In more cases than not we give that worn-out job a longer period of more perfect operation than it had before we put in pressure-proof rings.

Mr. Lítle cannot have made a study of pressure-proof rings. They are not difficult to install, they require no fitting, they do not break in use and service-stations do not report excessive breakage, although breakage of plain rings in installation and in use is common. It is possible to produce a plain one-piece ring with a smooth bearing surface and to fit it to the piston grooves carefully, provided the groove is not machined oversize, but this is a difficult operation in production jobs and one not likely to be obtained. After you have accomplished this difficult task to secure perfect operation, how long does this state of perfection last? It can be stated truthfully that with each mile the engine runs there is less perfection, while with a pressure-proof ring exactly the reverse is the result, as the ring will improve with use instead of deteriorate.

A MEMBER:—It seems to me that the best argument for the ring of narrow width has been missed. The thing that causes the groove to wear more than anything else is the force of the ring bearing against the groove at high speeds, due to the inertia of the ring, or its pressure against the side-walls; generally, the two operate together, causing the ring to slap from one side of the groove to the other. With a narrow ring, these forces are reduced very much more in proportion to the area of contact of the ring on the side grooves. From those considerations, the pressure caused on the side of the groove, due to the slapping of the ring in it, would be very much reduced per square inch; hence, the wear of the groove would be lessened to a very considerable extent. I believe also that the pumping of oil is a matter of inertia. Mr. Bull stated that an engine pumped more oil when the cylinder-head was removed than with the cylinder-head in place. The thing that is not taken into consideration is that the pressure of the gases on top of the piston-head and their attempt to seep past the ring during at least two strokes of the cycle reduces that oil-pumping considerably. When the cylinder-head is removed, there is no pressure on top of the piston; hence, the oil-pumping goes on at every stroke, instead of at every other stroke. I believe that accounts for this part of the difficulty. The result of our experience has been that we get better results from four narrow rings than we do from three wide ones. They stay in the groove longer and last very much longer in the engine. I believe that the matter of workmanship on the ring has more to do with its operation than does the particular design. If the ring fits the groove properly at the start, there cannot be a hammer-blow on it because it has no space to move in until it begins to cut a space.

MR. MANES:—The one-piece ring cannot be fitted very tight in the groove. It must have reciprocating space when the piston is cold, and that allows the clearance in the groove to increase more rapidly than if the ring were fitted closely. It is true that if the ring is fitted carefully, has deep enough grooves and a good wearing surface, it will wear a long time. In former days we had no trouble until an engine was 5 or 6 years old; but we had hand-fitted rings at that time. We had no ⅛-in. groove in a 3½-in. piston. We used a deep groove and the results told; but, with the present method of quantity production and the careless way in which the rings are installed in the piston, we certainly get rapid groove-wear and we pay for it by buying oil for our engines. We make the grooves apparently at random today, in most pistons, and that is where the trouble lies.

Mr. Damon:—Under some conditions, we use two-piece rings of the pressure-proof type and under other conditions, with the water-cooled engine, for instance, we use the 1/8-in. ring. We find we can get good results from both; but, in the majority of cases, we find that the two-piece ring will last longer and give more economical service for a greater length of time than a ring that is not ground to fit the groove. The practice we maintain is to equip every piston that is sent out with rings ground to fit the groove. We have yet to find a single-piece ring that we do not need to buy oversize and grind down to fit the grooves, on account of the warping that Mr. Bull mentioned.

J. E. Diamond:—Mr. Bull speaks of the friction set up by what are erroneously termed slotted pistons. In the partial circumferential separation of the head and the skirt resides the real merit of any and all of these newer types of aluminum pistons. As a matter of fact, in the case of at least one type of free-skirt piston the theory of differential expansion mentioned by Mr. Bull and featured in the elliptically machined piston was utilized a year in advance of even the conception of the latter. In this particular case, however, in addition to whatever benefit there might be derived from this differential expansion, was the suitable provision of circumferential skirt-expansion with mechanical construction that must be accepted. Mr. Bull's experience has been happy in comparison with others who have had an intimate association with the so-called constant-clearance type, which designation is a misnomer because the clearance is not constant.

I have been doing considerable work with one company employing this type, a company turning out very accurate work. Invariably, after every block test it is necessary to touch up each individual piston in varying degrees, and in many cases this operation is necessitated after a second or third test. In no two cases do the pistons show up the same. This surely is not particularly commercial. Furthermore, these pistons show excessive wear in service, and replacement is necessary almost invariably between 10,000 and 15,000 miles, which is much too frequent. A piston of the type advocated by Mr. Bull certainly does not have universal application so far as we have gone with it.

In connection with the determination of temperature, it was my intention merely to indicate the attempts that were made in the early days. Anent Mr. Jehle's remarks, it is the piston-ring that turns the trick with reference to the dissipation of heat. I am familiar with a series of experiments with a ring of unusual construction showing indisputably how important the bearing ring is in the matter of absorbing the heat from the piston-head and transferring it to the cylinder-wall. Circumferential slot or slots have very little or nothing to do with the matter of heat interference.

PISTON-RINGS[1]

By John Magee[2]

The author believes the piston-ring problem to be an engineering one worthy of serious study and that it should be possible to standardize types and sizes in a way that will go far toward elminating present difficulties.

It is stated that cast iron is the only satisfactory metal suitable for use in the internal-combustion engine and that the foundry offers the greatest opportunity for improvement, in the elimination of poor castings. The superiority of individually cast rings is averred and a formula for their composition is given.

Leakage and oil-pumping are discussed, followed by comment upon the width and form most desirable for piston-rings; and some of the difficulties of their manufacture are enumerated, together with suggested improvements, inclusive of inspection and testing methods.

Since many piston-rings have failed in the performance of their function either through faulty engineering or careless manufacture, piston-rings have developed into a problem. It seems that the problem is logically an engineering one, that piston-rings should offer a real opportunity for study and that it should be possible to standardize types and sizes in a way that will go far toward eliminating present difficulties. With this in mind, any discussion covering practices of today should assist in formulating a consensus of opinion as to the best practice.

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[1] Detroit Section paper.

[2] Detroit Piston Ring Co., Detroit.

While bronze, Swedish iron and even malleable iron and steel have been tried, it will be conceded that, so far, cast iron is the only satisfactory metal suitable for piston-ring usage in the internal-combustion engine. The density, the resiliency and the small cross-sectional area each being an important factor, it is evident at once that the foundry offers the greatest opportunity for improvement toward piston-ring perfection. Manifestly, with poor castings at the start, very little better than poor results can be expected at the finish. There is little question as to the superiority of the individually cast over the pot-cast piston-ring. Table 1 gives a mixture formula for individually cast rings.

Extreme care in the selection of materials, combined with frequent physical tests, will be necessary to maintain the standard. A required property of a test-bar ½ in. square is a Shore hardness of 35 to 40 or a Brinell hardness of 200 to 230.

TABLE 1—FORMULA FOR INDIVIDUALLY CAST PISTON-RINGS

Substance	Per Cent
Silicon	2.50 to 3.00
Sulphur, maximum	.... 0.70
Phosphorus	0.30 to 0.50
Manganese	0.45 to 0.70
Combined Carbon	0.50 to 0.60
Graphitic Carbon	2.75 to 2.65

LEAKAGE AND OIL-PUMPING

Many of the large number of piston-ring designs include some special joint. Because the periphery of a piston-ring is broken only at the joint, the impression seems to have prevailed that most leaks could be located there. Some of the devices for effectually sealing the joint are real tributes to inventive genius. As a matter of fact, the joint occupies such a relatively small percentage of the circumference that its effect on the whole is rather insignificant. In other words, it would be possible to construct a perfectly sealed ring-joint and yet have about 98 per cent of the remainder of the circumference leaking gas and causing loss of compression-pressure.

R. E. Lawrence, professor of mechanical engineering at the University of Detroit, recently calculated the leakage using a formula for the flow of gas through an orifice. The dimensions taken were those of the Ford-engine cylinder, with 0.002-in. clearance between the piston and the cylinder wall. Both tangs of a step-cut piston-ring were removed, and 3/16 in. on the circumference of the ring was open. If the piston were then held still during a period equal to the duration of a stroke at a 15-m.p.h. road speed, a pressure of 450 lb. per sq. in. would allow 0.0006 cu. ft. of gas to pass through the opening. This represents, by this calculation, the maximum leakage through a joint under the most "favorable" conditions. Such conditions never actually exist in the cylinder. The explosion rarely generates a pressure of 450 lb. per sq. in. and in any event the pressure always drops very quickly during the power stroke. Then, if there are three rings instead of one, with a film of lubricating oil helping to impede the progress of the escaping pressure, it will be seen readily that the actual amount of gas which could pass through the joint of a piston-ring under ordinary conditions would be expressed in a fraction much smaller than 0.0006 cu. ft.

In regard to so-called oil-pumping, it seems advisable to keep the ring periphery as nearly continuous as possible. The step-cut provides a joint that accomplishes this well and makes it the most desirable for all purposes.

Width and Form

The proper width of piston-rings is another subject productive of much discussion. The reduction in weight of all reciprocating parts is certainly to be desired. When it is taken into consideration that any added weight on a piston-ring must be multiplied by a factor of 6 to determine its equivalent inertia effect for each cylinder, the advisability of narrow widths is evident immediately. Theoretically a knife-edge in contact with the cylinder wall will produce the proper result. It remains only to establish the added width necessary to allow for practical production, always considering that the minimum width is desired. Present methods of manufacture indicate that ⅛ in. is the proper width, all things considered. The thickness of the ring or the depth of the groove is subject to the same consideration in all ways.

The merits of both eccentric and concentric forms have been discussed from time to time. No doubt, the eccentric ring is more correct for theoretical uniform wall-pressure. However, if the pattern for the casting is designed for the ring at its full opening, and the natural surface density of the inside of the ring is left undisturbed in machining, a proper foundry-mixture will produce a concentric ring with a wall pressure that is so nearly uniform in actual operation that its many other advantages make it preferable. It should be remembered also that the theoretical eccentric ring tapers down from its heaviest section, opposite the joint, to a knife-edge at the joint. Any design for an eccentric ring must modify this form to a certain extent, to avoid the thin wall at the joint.

MANUFACTURING DIFFICULTIES

It is not my intention to enlarge upon the proper methods of manufacture of piston-rings. It must be admitted, however, that more failures of piston-rings are due to faulty manufacturing than to faulty engineering principles. After all, from an engineering standpoint, the function of a piston-ring is a simple one. Unfortunately, the manufacturing problem is not so simple as is supposed sometimes. Some study of the manufacture of piston-rings will facilitate developing specifications that are likely to be met.

All piston-ring manufacturers prefer to make rings with the finished surfaces ground to size. It is easier to hold accurate dimensions on a grinding machine than it is on a lathe. However, hard castings and hard spots in castings are machined without difficulty or detection by grinding. For this reason piston-rings with a *turned* finish on the diameter dimension should be specified. Neither the scleroscope nor the Brinell-hardness test will disclose hard spots in castings. A ground finish may cover up many hard spots or a hard scale. The production of a turned surface is a somewhat slower process for the manufacturer, but it guarantees a uniform soft wearing surface.

Flatness is very essential, since a serpentine condition will allow leakage around the back of the ring through the ring groove. If all of the internal stresses are not removed during the process of machining, the ring is as apt to warp sidewise as it is to become elliptical. Therefore, width dimensions should be inspected with a light gage instead of with a micrometer. With a ring lying on a perfectly flat surface, a side-warp will cause it to register oversize. With measurements at intervals, the two points of a micrometer might indicate a parallel width on a ring considerably warped.

The most common "defect" in the manufacture of piston-rings is the elliptical ring, generally termed out-of-round. The specified out-of-round tolerance may vary according to the amount that the ring is likely to be worn-in on the block. At best it is desirable to keep the variation within very low limits, say 0.00025 to 0.00050 in. If the tolerance is expressed in light-gage terms, it is much more simple for inspection, because the light-gage is at present the most practical inspection device for locating out-of-round rings.

No accurate data are available for determining the poundage or wall pressure of piston-rings to accomplish definite purposes. Actual experiments, conducted at different times for different purposes, seem to indicate that a poundage in excess of 4 lb. per sq. in. of bearing surface is needed to prevent collapse under pressure. Therefore, a poundage of 5 lb. per sq. in. has commonly been specified for all purposes.

There seems to be a diversity of opinion as to the proper gap-opening or expansion allowance. This probably is due to the difference in the estimated temperatures to which a piston-ring is subjected. Using 0.0000056 per in. per deg. as the coefficient of cast-iron expansion, a minimum opening-allowance for maximum expansion is obtained by multiplying this coefficient by the circumference of the ring.

The foregoing comments are made with a view to the standardization of the best engineering and manufacturing practice, rather than to attempt to dictate the ultimate design for piston-rings. The ever-increasing number of sizes and designs, some of which are freakish, is certainly an indication of the dire need for such a standard.

[The discussion of this paper is printed on p. 296.]

ADVANTAGES OF LIGHT-WEIGHT RECIPROCATING PARTS[1]

By L H Pomeroy[2]

After pointing out that the general question of weight reduction is no exception to the fallacies that seem to have beset the development of the automobile from its earliest days, the author outlines briefly the problem confronting the automobile designer. The influence of the weight of the reciprocating parts on the chassis in general and the engine in particular is emphasized as being of greater importance than the actual saving in the weight of the parts themselves, it being brought out that the bearing loading due to inertia is really the factor that limits the maximum engine speed. Reference is made to the mathematical investigation by Lanchester in 1907 of the advantages of using materials of high specific-strength and the conclusions arrived at are quoted in full. A tabulation of the specific strengths of various materials used in automotive engineering practice is presented as showing the advantages of aluminum as compared with steel.

The savings in weight that are possible by use of aluminum without any sacrifice of strength are next pointed out. Comparison is made of the stiffness of steel and aluminum sheets as a specific instance of weight reduction and this is followed by an extended consideration of aluminum connecting-rods, including an analysis of the loading due to inertia throughout a complete four-stroke cycle, and a comparison of steel and aluminum connecting-rods on a weight basis. The advantages of using aluminum to secure the required stiffness in a connecting-rod because of its low density are emphasized, it being brought out as the result of a mathematical analysis that equal stiffness as compared with steel can be secured in an aluminum connecting-rod with about one-half the weight of the material. An extended comparison of steel and aluminum connecting-rods that have been in service is next presented. The production methods employed for steel connecting-rods are stated as being applicable to aluminum. The advantages of the combination of the aluminum piston and the connecting-rod are pointed out, it being stated that a saving of 15 lb. in this connection as compared with a cast-iron piston and a steel connecting-rod re-

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[1] Buffalo Section paper.

[2] M.S.A.E.—Consulting engineer, Cleveland.

sults in an overall saving of about 14 times this amount.

Although automotive engineering science is now arriving at the stage where there are few engineers who claim that weight is advantageous apart from the necessities of strength, the general question of weight reduction is by no means free from the fallacies and false theorizing that seem to have beset the development of the automobile from its earliest days. For example, the average salesman will assert with no little emphasis that it must require more energy to reciprocate a heavy piston than a light one and that the horsepower of the engine is correspondingly affected. This may or may not be true according to the design of the pistons and their relative friction, for it is very easy to conceive of a tight-fitting light-weight piston with many rings offering a greater resistance to motion than a slack-fitting piston of twice the weight and say one ring. The point is, of course, that if any mass is put into motion and afterward brought to rest as in the case of a piston during its travel, no work is done apart from friction. In other words, the energy put into the piston to start it moving is given up by it during its period of slowing-down to rest. With even greater emphasis it is claimed that the use of light pistons reduces vibration. This again while true as a general proposition is by no means an immediate truism. Most engines run at their best when the inertia-pressure diagram is approximately midway between the compression and expansion lines of the gas-pressure diagram, and it is again easily conceivable that for an engine running at a constant speed increasing the weight of the piston might improve engine smoothness.

The case for light reciprocating parts, however, rests upon grounds which are overwhelmingly more important than the somewhat hair-splitting considerations mentioned above. Before presenting this in detail it is of interest to review briefly the problem before the automobile designer. With a Pierce-Arrow at one end of the scale and a Ford at the other it is possible only to generalize vaguely but there are certain things in common: first, an approximately equal passenger carrying capacity, although the seven-passenger Pierce-Arrow is usually occupied by three or four persons in luxury, while the five-passenger Ford is usually occupied by seven or more in acute discomfort; second, the capacity to traverse any road upon which the wheels can hold; and third, the maintenance with safety of at least the legal limit of speed. These three items can, of course, be supplemented but they cover more or less those chiefly related to road performance.

We have then in an automobile a passenger load supported upon a frame, axles and wheels, together with an engine for propulsive purposes. From the viewpoint of the total weight involved it is obvious that the passenger weight is in accordance with the dictates of birth and diet and not under the control of the designer. The remaining weight is determined chiefly by the selling price of the car, and the problem resolves itself into giving the public the maximum aggregation of virtues that will result in that combination of sales on the one hand and profit on the other, essential to commercial success and stability. This may be differently expressed by saying that the cost of material is the largest single item of the three factors of cost, namely, labor, overhead and material, and that any reduction of the material cost, that is any reduction of the weight and the dimensions, goes hand-in-hand with the reduction of labor and consequently of overhead charges.

As in everything else the more one pays the more one gets or at least expects to get. The man who buys a typical heavy car does so because he associates with such a car the cardinal virtues of reliability, comfort and first-class road performance and, as it is difficult to obtain these in any other way, he does not resent the relatively high running-costs involved.

It is, therefore, the business of the engineer to fulfil these conditions by applying his knowledge of engineering science and research to the reduction of weight and thereby of running costs without sacrificing one iota of reliability, luxury or longevity, and at the same time to reduce the cost of production so that engineering and economic ideals will progress together.

An examination of the components of an automobile indicates that the bearing surfaces and the weight are closely interrelated and that in orthodox construction at any rate it is possible to build down to a given weight only by reducing the bearing surfaces to the minimum. Throughout an automobile we find this holding true; a few examples are taken at random for purposes of illustration. In the engine the crankshaft dimensions control its own weight and the weight of the bearings supporting it; similarly with the camshaft, pistons, wrist-pin, tappets and other parts. The weight of the clutch is determined by the area of the friction contact; and that of the transmission by the dimensions of the gear teeth and centers, the desired stiffness of the gear shafts and the permissible load upon their bearings, which items in turn decide the dimensions of the case that contains them. The same remarks apply to the universal-joints, the rear-axle bevel and differential gears, the front and rear hubs, the brakes and the steering-gear. The essential difference between the heavy and the light car is in the difference between the factors of bearing wear and the rigidity of construction appropriate to the problem in hand.

The load upon any bearing is due in part to the weight arising from its own dimensions in order that it may be adequately supported and in part to the passenger and body load supported by the chassis as a whole and the power requirements thereof. For approximately the same passenger carrying capacity and body accommodation it is easily possible to have a variation in car weight of from 2400 to 5000 lb. Allowing that the weight of the body fitted to the lighter car is say 700 lb. and that on the heavy car it is say 1100 lb., the chassis weight of the light car becomes 1700 lb. and that of the heavy car 3900 lb., a striking example of how ideals differ in producing two articles professing to do approximately the same job. Actually, of course, they do not do the same job, if the average light car were put to do the work possible with the heavy car, it would give up the ghost very early. The larger shaft dimensions, bearing surfaces and supports in the heavy car have been shown by high-duty experience to be necessary.

While admitting and even claiming this, there have been developments during the past few years in the field of light alloys which profoundly modify the whole problem of automobile design and make it perfectly demonstrable that a wholesale reduction in the weight can be obtained with the same or even higher factors of safety and wear in the bearings, and without the slightest sacrifice of the stiffness of shafts and general chassis construction so essential to a car that is required to meet the most exacting conditions. The object of this paper is to show more particularly the advantages to be derived by attacking one part of this problem only, namely the reduction in the weight of the reciprocating parts and to leave to the imagination the possibilities that can be achieved when this is coupled with the general weight reduction referred to. As will be seen the weight of the reciprocating parts has an influence upon the weight of the chassis in general and the engine in particular, which is vastly more important than the weight saved in the parts themselves.

The extent to which engine dimensions are a function of inertia rather than of gaseous pressures is often overlooked. At very high speeds and part throttle, as when driving downhill, the inertia pressures can easily be much greater than those due to the explosion. On the other hand, when at full throttle and low speed, as when pulling on high gear uphill, the situation is reversed. It becomes of interest and importance, therefore, to obtain some approximate idea of the conditions under which the inertia forces are greater than those due to the explosion.

The capacity of a bearing to withstand wear between the limits of the oil being crushed out of the bearing due on the one hand to heavy pressure and on the other to being evaporated out by the heat generated at high speed, is measured by the product of its mean loading in pounds per square inch and its peripheral velocity in feet per second. This value in good automobile practice should not exceed 16,000. At very low engine speeds the inertia forces are negligible compared to the gaseous pressurees, so that for a crankshaft say 2¼ in. in diameter running at 400 r.p.m., or a peripheral speed of 4.7 ft. per sec., the permissible limit pressure would for a load factor of 16,000 be some 3400 lb. per sq. in., which is very much greater than that actually arising under such conditions. At high speeds, say 2800 r.p.m., with a peripheral speed of 32.9 ft. per sec. the allowable unit pressure would be some 485 lb. per sq. in., a value frequently attained and even exceeded in existing automobile engines.

It may be said that if the bearings of an automobile engine are designed to take care of maximum-speed conditions, low-speed conditions will take care of themselves. As suggested, many automobile engines are now running at speeds that are up to and in some cases above those permissible for bearing reliability and it is not too much to say that bearing loading due to inertia constitutes the real upper limit of commercially possible engine speeds.

The advantages of the use of material of high specific strength, or strength per unit weight, for reciprocating parts were mathematically investigated by Lanchester[3] in 1907, but like many investigations this was somewhat ahead of its time. His generalizations are much more important and applicable to engine design today than they were when they were written 15 years ago, and constitute a good example of how pure mathematical reasoning from fundamentals finds a definite application when empirical developments have cleared the way for them. He pointed out that the limiting speed of engines is determined by the strength of materials and that if similar engines be compared it is possible to predict the relative safe speeds at which they can be run.

The gist of these conclusions is quoted below and the engineer is recommended to study the paper in full.

INFLUENCE OF CHANGES IN THE DENSITY AND STRESS ON THE HORSEPOWER DEVELOPED

> We will now revert to the general expression
>
> $$\text{Hp} = (\sigma^{1.5}/\rho^{0.5})\ l^2 \times \text{a constant}$$
>
> and discuss the influence of changes in the physical attributes of the materials employed, i.e. variations of σ and ρ (stress and density).
>
> Translated into ordinary language the expression shows that in similarly designed engines the horsepower varies as the 1.5th power, that is, as the cube of the square root of the stress, and as the square root of the density of the materials employed.
>
> Now it is evident that the weight of the engine also will depend upon the variables and l, and for the conditions of geometrical similarity the form of this expression is
>
> $$W = \rho l^3 \times \text{a constant}$$
>
> so that the horsepower per unit weight, which is the quantity of most interest to us, will be
>
> $$\begin{aligned}\text{Hp}/W &= [\sigma^{1.5}/(\rho \times \rho^{0.5})] \times (l^2/l^3)\\ &= [(\sigma/\rho)^{1.5} \div l] \times \text{a constant}\end{aligned}$$
>
> Let us denote the quantity σ/ρ by the symbol Φ, and term it the "specific strength" of the material; then we have
>
> $$\text{Hp}/W = \Phi^{1.5}/l$$
>
> We have now the question of weight saving in a nutshell. The above expression shows that to which I have already drawn your attention, the importance of subdividing the power unit by employing a multiplicity of cylinders of individually small size, for we have the horsepower per unit weight inversely as the linear

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[3] See *Proceedings of the Institution of Automobile Engineers*, vol. 1, p. 155.

dimension, the latter, l, being the denominator in the above expression. We can also see at once the importance of employing materials of high specific strength; the form of the expression shows that if we can, by employing all-round a higher grade of material, say of 10-per cent greater specific strength, we shall effect a saving of weight of approximately 15 per cent.

Of course, it is not always possible to effect an improvement in the quality of the material in every part of a machine, and it is of considerable interest to us to ascertain where and how the saving in weight is most usefully effected.

WEIGHT SAVING CONSIDERED IN DETAIL

Let us, to fix our ideas, suppose that we have at our command two kinds of material, one of which has just four times the specific strength of the other; and let two carefully designed engines be built to the same specification, one from each kind of material. Now it is evident that, part for part, the one engine can be built one-fourth the weight of the other. There may be some slight difficulties in design, owing to the slenderness of some of the parts, but we can brush this difficulty to one side by supposing the difference of specific strength to be wholly due to a 4 to 1 difference of density, that is, σ remains constant.

So far we have accounted for the Hp/W varying in the direct ratio of Φ only, but the one engine will not only be lighter than the other but it will develop more power, for its reciprocating parts will give rise to less inertia and the revolution speed can be increased. The extent to which the revolution speed can be increased is in the inverse ratio of the square root of the weight of the parts, or in the case in point the revolution speed can be doubled. Thus the horsepower of the lighter engine will become twice as great as that of the heavier one, or its Hp/W will be 4×2; that is, eight times as great, which is $4^{1.5}$ in accordance with the equation.

We thus see that on the former supposition of a 10-per cent improvement in the material, producing approximately a 15-per cent improvement in the power weight factor, 10 per cent of this improvement is due to the direct lightening of the engine and 5 per cent to the increased power derived from the higher revolution speed rendered possible.

It is thus evident that by far the greater importance attaches, relatively speaking, to the quality of the material employed in the pistons and connecting-rods, for these reciprocating parts do not usually exceed 10 per cent of the total weight of the engine, and attention given to this 10 per cent is of as much effect as similar attention devoted to any other 50 per cent of the engine. It is thus found advantageous to adopt the very highest class of material for pistons and connecting-rods. For some years past I have employed a high grade of nickel steel both for the connecting-rod stampings and for the blanks from which the pistons are turned and I believe that the results would justify even more attention still being paid to the reduction of weight in these organs.

A SECONDARY EFFECT

A secondary effect, which must not be lost sight of, results in a saving of weight which is not obvious from a mere inspection of equation for Hp/W.

We have seen that the change in the power-weight factor as due to $\Phi^{1.5}$ takes the form of a saving of weight in the direct ratio of Φ, and in an increase of power in the relation $\Phi^{0.5}$. But we may not want increased power; it is usually some stated power that is required, so that l^2 will require to vary inversely as $\Phi^{0.5}$, that is, l varies as $l/\Phi^{0.25}$. Substituting, we have

$$\text{Hp}/W \propto \Phi^{1.75}$$

under the conditions of stated horsepower, that is to say, $\text{Hp} = \text{a constant}$. This may be expressed alternatively by saying that for a given horsepower, for an engine of given numbers of cylinders, the weight varies inversely as $\Phi^{1.75}$.

The first equation may be written in the form

$$\text{Hp} = \sigma \times \Phi^{0.5} \times l^2 \times \text{a constant}$$

In this form the σ relates to the stress in the working fluid that is the cylinder pressure; taking this as constant we have

$$\text{Hp} \propto \Phi^{0.5}\, l^2$$

and when Hp is constant we have

$$\Phi^{0.5} \times l^2 = \text{a constant}$$

or

$$l \propto (1/\Phi^{0.25})$$

which gives the same result as before,

$$W \propto (1/\Phi^{1.75})$$

We thus see that the saving of weight to be effected by employing high-grade material is even more than we had hitherto concluded, so that a 10-per cent higher specific strength would give about 17.5 per cent, instead of 15 per cent as previously concluded. The earlier figure was perfectly correct so long as the linear dimension of the engine was the constant, instead of the horsepower.

USE OF ALUMINUM

In the searching that has occurred since the facts were recognized for materials in which the tensile-strength was high per unit weight, the manifest advantages of aluminum as compared to steel have been overlooked or not taken seriously.

The specific strength of the various materials commonly used in automotive engineering practice rank as given in Table 1.

From Table 1 it will be seen that the specific strength of forged aluminum closely approaches that of a 3-per cent heat-treated nickel-steel, while cast aluminum is greatly superior in specific strength to any other cast material in common use. It follows, therefore, that by suitably increasing the dimensions of a part previously made in any of the other materials mentioned, except 3-per cent nickel-steel, it can be made in aluminum to give the same strength but to weigh considerably less. This argument applies directly to cases of pure stress, tension, compression and shearing. There is, however, another very important aspect of the case, which in practice annihilates the superior specific strength of the high-alloy steel, arising from the fact that where compound stresses are involved dimensions per se confer strength.

TABLE 1—SPECIFIC STRENGTH OF AUTOMOTIVE MATERIALS

	Tensile-Strength, Lb. per Sq. In.	Weight per Cubic Foot, Lb.	Specific Strength
Forged Aluminum	60,000	180	332
0.20-Per Cent Carbon-Steel	80,000	490	163
0.35-Per Cent Carbon-Steel	105,000	490	212
3-Per Cent Nickel-Steel	170,000	490	348
Sand-Cast Heat-Treated Aluinum	30,000	180	161
Chill-Cast Heat-Treated Aluinum	40,000	180	212
Malleable Iron	45,000	480	94
Steel Castings	60,000	480	125
Hard Cast Bronze	35,000	540	65
Cast Manganese	60,000	540	112

It is well known that in engineering design generally examples of pure stress are conspicuous by their absence. Even in the simple case of a nut and bolt nominally in tension, it is very doubtful if the nut can be tightened without involving some degree of bending due to the thread being non-axial.

A simple example of compound stress is that of a rectangular beam in which doubling the depth quadruples the load carrying capacity. It may be urged that this applies to all materials, as in truth it does, but a very large portion of the section in the region of the neutral axis of a beam is only lightly stressed compared to that at the top and bottom. This constitutes useless material and it is obvious that if the necessities of design compel useless material to be carried, as they often do, the lower its specific gravity the greater is the weight saving effected. It can, of course, be argued against this that it is possible to design the section of a beam so that this useless material will be removed by the processes of manufacture, as in the case of rolled steel sections. Unfortunately, the processes of manufacture to attain the end of eliminating the useless material are limited in scope and expensive in application and this fact constitutes an exceedingly important claim for the engineering and economic advantages of the broadcast use of aluminum in its various forms.

The essence of the technique of weight saving is a study of stress distribution and the design of parts so that the material used is proportional to the load at any point. While it is practically impossible to attain this end, at any rate in most automobile parts, due to the limitations of fabricating processes, the best alternative is to use a material in which the useless portion is of minimum weight.

In automobile design, however, the question of the strength of the various parts is but one aspect of the problem. The majority of parts need consideration from the point of view of stiffness rather than strength.

The history of automobile design is that of the increase in the dimensions of important details to overcome vibration and whippiness. For example, automobile frames are now girders of very great strength, but the strength is entirely secondary to the fact that stiffness is the dominant consideration in the design of a frame so that a closed body can be mounted with a reasonable chance of being able to open or close the doors after 6 months of use. Similarly with axles, transmission cases, crankcases, crankshafts, and other parts, while these were found strong enough in ancient designs they were not stiff enough.

The strength of a beam, nearly every part of an automobile is a beam in one sense or another, is a function of the square of its depth and its safe tensile-stress, while its stiffness is a function of the cube of its depth and its modulus of elasticity. As this last expression is so intimately connected with the application of aluminum to engineering design it may be worthwhile to explain that the modulus of elasticity of any substance is the ratio of stress to strain within the elastic-limit, the stress being the load per square inch and the strain the extension or compression in inches caused thereby divided by the original length of the piece. In other words, the modulus of elasticity of a material is the load that would double the length of a bar of the material 1 sq. in. in section if the section remained constant. For the common engineering materials, such as steel, cast iron and bronze, the moduli of elasticity are approximately 30,000,000, 17,000,000 and 14,000,000 lb. per sq. in. respectively, while for aluminum it is about 10,500,000 lb. per sq. in. It is of interest to note that the modulus of elasticity is a function of the character of the material rather than of its precise analysis or tensile-strength. Thus low-carbon steel and the nickel-chrome steel-alloys that may vary in tensile-strength by hundreds of per cent do not vary 10 per cent in the modulus of elasticity. Similarly the bronzes and aluminum alloys take their modulus from their basic material and are relatively little affected by the materials that compose the various alloys.

STEEL AND ALUMINUM SHEETS COMPARED

With the above in mind it is of interest to compare the stiffness of a sheet of steel with that of ordinary rolled aluminum. For similar supporting means, as in the panel of a door, for example, the deflection due to a load applied at any similarly situated point in each sheet is inversely proportional to the modulus of elasticity and the cube of the thickness. If the thickness of the steel sheet is say 0.04 in. and that of the aluminum 0.06 in., the relative deflections will be as

$$1/[(0.04)^3 \times 30{,}000{,}000] : 1/[(0.06)^3 \times 10{,}500{,}000]$$

or as

$$[(0.06)^3 \times 10{,}500{,}000] \div [(0.04)^3 \times 30{,}000{,}000] = 1.18 : 1$$

The aluminum sheet is, therefore, 18 per cent stiffer than the steel sheet and 50 per cent thicker. As the weight of a steel sheet 0.04 in. thick is approximately 1.6 lb. per sq. ft., while that of an aluminum sheet 0.06 in. thick is approximately 0.9 lb. per sq. ft., it will be seen that this *increase* in stiffness of 18 per cent is accompanied by a decrease in weight of 0.7 lb. per sq. ft., or 43 per cent.

The relative strengths in the above example depend upon the square of the thickness and the ultimate tensile-strength, so that if the steel sheet has an ultimate tensile-strength of 55,000 lb. per sq. in. and the aluminum 25,000 lb. per sq. in., the relative strengths are as $55,000 \times 0.04^2$ to $25,000 \times 0.06^2$, or as 90 to 88 in favor of the aluminum. With heat-treated aluminum sheet the advantage of aluminum in respect of strength compared to steel would be in the order of 2.4 to 1. This is a perfectly legitimate example of the weight saving that can be obtained by using aluminum in one of its simplest applications to the automotive industry. A survey of fabricating methods and of other metals in general use fails to suggest any other way of obtaining a given desired strength and stiffness with such reduction in weight.

It must be emphasized that by no means the least important aspect of the case for aluminum as a means of obtaining light-weight construction without any sacrifice of strength or stiffness is that practically all engineering design tolerates a vast waste of material in the interest of economical fabrication. For example, a brake-rod that is threaded at each end has only the strength of the metal at the base of the thread, so that all areas in excess of this are useless. Similarly, a rolled steel beam supporting a floor has a constant cross-section instead of one proportional to the bending-moment applied.

With light-weight alloys the same conditions arise and there is of course in similar designs the same percentage of waste materials. The point, however, is that the absolute dead-weight of useless material in the region of the neutral axis is of far less consequence.

THE ALUMINUM CONNECTING-ROD

The enunciation of these general principles leads to the particular consideration of the aluminum connecting-rod. From the point of view of specific strength, it has been shown that forged aluminum is greatly superior to the steels used in the vast bulk of automotive engine connecting-rods where from the *strength* point of view a steel giving from 80,000 to 90,000 lb. ultimate tensile-strength has proved perfectly satisfactory. There is no doubt that the development of the forged aluminum connecting-rod follows logically from the aluminum piston, to which many makers have been forced to return, and for the same reason, the reduction of internal wear-and-tear in engines by 50 per cent, and the vastly improved performance obtained thereby.

The primary advantage of forged aluminum as a material for high-duty connecting-rods in high-speed internal-combustion engines arises from the fact that the chief cause of bearing wear and failure is the loading imposed upon the crankshaft and connecting-rod bearings due to the inertia of the moving parts.

The loading on the connecting-rod big-end bearings is composed of two parts; (*a*) that due to the fluid, the gaseous mixture in the cylinder, pressures during compression and expansion, and (*b*) that due to the inertia forces arising solely from the mass of the moving parts, the piston and the connecting-rod.

If the four strokes of the conventional four-stroke cycle are examined the following will be apparent:

(1) At the beginning of the induction stroke, the loading of the big-end and adjacent crankshaft main bearings is due to piston and connecting-rod inertia only and acts on the inner side of the crankpin, or that nearest the axis of the crankshaft

(2) Slightly before the middle of the induction stroke the piston inertia forces vanish but the full inertia of the connecting-rod big-end is exerted on the inner side of the crankpin

(3) At the end of the induction stroke the full effect of piston and connecting-rod inertia still is exerted on the inner side of the crankpin

(4) At the beginning of the induction stroke the full effects of piston and connecting-rod inertia still are exerted on the inner side of the crankpin

(5) At the middle of the compression stroke the piston inertia vanishes and the loading on crankpin is due to connecting-rod big-end inertia, the direction still being toward the inner side of the crankpin but slightly modified by the fluid pressure

(6) At the end of the compression stroke the loading on the crankpin is the difference between the fluid loading due to compression and the inertia pressure of piston and connecting-rod. Hence at the moment before ignition the crankpin is

subject to its smallest loading, at least at speeds of 1000 r.p.m.

(7) At the beginning of the explosion stroke the loading on the crankpin is due to the difference between the explosion pressure and the inertia pressure of the piston and the connecting-rod. At high speeds and part throttle these may also neutralize each other

(8) At the middle of the explosion stroke the preponderating load is due to the fluid pressure but the centrifugal loading due to big-end inertia remains

(9) At the end of the explosion stroke the loading of the crankpin is due to the sum of the inertia pressures due to the piston and the connecting-rod plus that due to the fluid pressure, and is exerted also on the inner side of the crankpin

(10) At the beginning of the exhaust stroke the loading is on the inner side of the crankpin and due to piston and connecting-rod inertia

(11) At the middle of the exhaust stroke the loading is on the inner side of crankpin and due to the big end of the connecting-rod inertia

(12) At the end of the exhaust stroke the loading is on the inner side of the crankpin, due to piston and connecting-rod inertia

With regard to the 12 positions of the crankpin mentioned, it will be seen that in three only, namely, the end of the compression stroke and the beginning and the middle of the explosion stroke, do the fluid pressures in any appreciable way counteract the effect of the inertia pressures. It will be seen also that the pressure on the inside of the crankpin bearing due to the centrifugal action of the big end of the connecting-rod is always present. This action is approximately that due to the weight of the big end of the connecting-rod, the weight obtained by placing the big end of the rod on a scale-pan, while the small end is freely supported in space. It should be noted also that at the top and the bottom of the stroke, the inertia pressure on the inner side of the crankpin is that arising from the whole mass concerned, or the complete piston *plus* the whole connecting-rod.

In general the inertia effect of the connecting-rod may be considered as if the weight of the big end as previously described were concentrated at the crankpin while the difference between this weight and that of the whole connecting-rod, or the weight of the small end, were concentrated at the wrist-pin. For example, a steel connecting-rod for a 3½-in. bore engine made as light as practicable weighs about 3.50 lb., of which 2.75 lb. can be reckoned as rotating mass and the remaining 0.75 lb. as reciprocating mass. In aluminum this could be reduced to 2 lb., of which about 1.5 lb. would be rotating and 0.5 lb. reciprocating mass.

A cast-iron piston for a 3.5-in. cylinder bore weighs about 2.25 lb. complete.

The effect of using an aluminum connecting-rod would in such a case reduce the big-end loading due to inertia, neglecting connecting-rod angularity, in the ratio of 2.75 to 1.50 at the middle of the stroke, where the piston inertia forces vanish, and in the ratio of 5.75 to 4.25 at the ends of the stroke, an advantage in the reduction of the load factor varying from 40 to 26 per cent, or a mean reduction of some 33 per cent.

The advantage thus obtained can be utilized by

(1) Reducing the width of the bearings by 33 per cent, which in turn reduces the overall length and the total weight of the engine
(2) Reducing the gear-ratio and running the engine at a higher speed, thus obtaining a better performance with the same factor of bearing safety
(3) Improving this factor in engines that are unduly supplied with bearing surface

In brief, the use of aluminum for connecting-rods affords a ready means of making great overall economies in a new design and of allowing a considerable development of existing designs.

Connecting-Rod Design

The greatest difficulty in connecting-rod design is to give adequate support to the babbitt or other material in direct contact with the crankpin without an undue increase of the weight. This is not a matter of strength of the supporting means, either steel or aluminum, but of securing stiffness. In this connection aluminum is particularly valuable owing to its low density. For example, a connecting-rod to suit a 2-in. diameter crankpin must be bored out to at least a 2⅛ in. diameter, leaving 1/16 in. for babbitt. The mean thickness of the metal surrounding the babbitt in steel should be at least ¼ in., neglecting big-end bolt bosses.

The mass of such a big end in steel is, therefore, proportional to the difference between the squares of the outside and the inside diameters multiplied by the density, or

$$[(2.625)^2 - (2.125)^2] \times 0.28 = 0.7$$

The stiffness for the same internal diameter is approximately proportional to the cube of the mean thickness times the modulus of elasticity or

$$0.25^3 \times 30,000,000 = 468,750$$

Since the stiffness of a part is proportional to its modulus of elasticity, other things being equal, and as the modulus of elasticity of aluminum is 10,000,000, while that of steel is 30,000,000, it follows that to obtain the same stiffness of the big end in aluminum as in steel the thickness must be increased accordingly.

$$\sqrt[3]{(30,000,000 \div 10,000,000)} = 1.44$$

The equivalent thickness, therefore, in the above example becomes

$$0.25 \times 1.44 = 0.36$$

From this the relative weight of the aluminum big-end is as before proportional to the difference of the squares of the outside and the inside diameters multiplied by the density, or

$$[(2.845)^2 - (2.125)^2] \times 0.1 = 0.36$$

Thus it will be seen that the same stiffness can be obtained in an aluminum rod for about one-half the weight of material required in a steel construction.

Apart from the amount of metal required to give the desired stiffness of bearing support, the question arises as to the number of bolts for securing the big-end bearing-cap. As a rule four bolts make a much better job than two and incidentally need be no heavier.

It is important to provide plenty of bearing surface between the cap and its abutment on the connecting-rod. The proportions of the small end of the connecting-rod are dependent upon the vagaries of the designer of the wristpin, but it should be kept in mind that the weakest part of the wristpin in a vertical engine is at the top of the rod, through which an oil-hole is generally drilled. The metal at this point should be 2½ to 3 times as thick as at the sides of the wristpin.

Coming now to the proportions of the shank of the connecting-rod, namely that part between the big and small ends, the region is entered in which much high-class mathematics may be applied. This would be justifiable if engineers had any complete knowledge of the distribution of the stress in a connecting-rod; as such knowledge is only now being acquired in respect of the very simplest structures, experience is still the best guide. In the nature of things it is highly probable that stress distribution is more uniform in a ductile metal such as forged aluminum than in less ductile metals such as nickel-chrome steel.

In the present state of knowledge it appears to be perfectly safe to make the section of an aluminum rod that replaces a reliable steel rod such that the section of the aluminum rod is similar to that of the steel rod and of twice the area. All designers will realize the number of exceptions there may be to this rule and the desirability of consultation with the prospective suppliers of the aluminum connecting-rod forgings.

The effect of aluminum connecting-rods on crankshaft bearing design is important. Many engine builders use counterbalanced crankshafts to reduce main-bearing wear, particularly on the middle main-bearing. There are, however, certain distinct objections to this practice. In the first place, a counterbalanced crankshaft is heavy and expensive, and in the second place it introduces a distinct liability to torsional periodicity, particularly in six-cylinder engines.

The necessity for such crankshafts arose from the bearing wear that was due in turn to heavy pistons and connecting-rods. The use of aluminum rods reduces very considerably the crankshaft skipping-rope action which causes bearing wear. It is safe to say that in combination with aluminum pistons as well, counterbalanced crankshafts are unwarranted and disadvantageous.

COMPARISON OF STEEL AND ALUMINUM CONNECTING-RODS IN PRACTICE

An interesting example of the truth of the foregoing remarks is obtained by comparing the characteristics of a steel connecting-rod taken from one of the best four-cylinder engines produced and of an aluminum rod that has done many thousands of miles under most exacting conditions without a suspicion of failure. The only criticism that can be urged against the steel connecting-rod under discussion is that it is too light, particularly in respect of the metal supporting the babbitted shell in the big end. The steel connecting-rod is from an engine of 3⅜-in. bore and 5-in. stroke; the aluminum rod is from an engine of 4⅛-in. bore and 4¼-in. stroke.

The detail dimensions and weights are contained in Table 2.

TABLE 2—COMPARATIVE DIMENSIONS OF STEEL AND ALUMINUM CONNECTING-RODS

	Steel	*Aluminum*
Length betweeen Centers, in.	12⅛	11⅝
Diameter of Big-End Bearing, in.	1⅞	2
Length of Big-End Bearing, in.	2¼	2¼
Diameter of Wrist-Pin Bearing, in.	⅞	1⅛
Length of Wrist-Pin Bearing, in.	1¼	1⅝
Total Weight of Rod, lb.	2.900	2.140
Weight of Big End, lb.	2.340	1.690
Weight of Small End, lb.	0.560	0.450
Weight of Piston[4], including Wrist-Pin and Piston-Rings[5], lb.	1.206	2.040
Total Reciprocating Weight, lb.	1.766	2.490
Total Reciprocating Mass, lb.	2.340	1.690

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[4] Piston is made of aluminum in each case.

[5] The piston used with the steel connecting-rod has three piston-rings, while that used with the aluminum connecting-rod has four.

Now as previously stated the loading of the crankpin bearing due to the inertia forces is ascribable in part to the centrifugal effects of the rotating mass of the big end of the connecting-rod itself and in part to the reciprocating inertia-effects of the piston and the connecting-rod small-end. These latter apply at the top and the bottom of the stroke only, while the former acts continuously. The mean loading, therefore, is that due to the rotating mass of the big end plus *half* that due to the reciprocating masses, as these are fully manifested only at each end of the stroke and vanish about the middle thereof.

The actual pressures manifested are directly proportional to the weight of the parts in question, the square of the number of revolutions per minute and the stroke. In comparing the two connecting-rods, however, the speed of the engine may be neglected, as the comparison between the two rods at any engine speed will hold at any other. We have then the average loading of crankpin proportional to the stroke multiplied by the sum of one-half the weight of the reciprocating parts and the weight of the rotating parts.

For the steel rod this becomes

$$5\,[(1.766 \div 2) + 2.340] = 16.100$$

while for the aluminum rod the loading is proportional to

$$4.25\,[(2.49 \div 2) + 1.69] = 12.50$$

The advantage obtained in reduction of big-end bearing pressures by the use of aluminum connecting-rods is thus clearly manifest, especially when it is noted that in each case aluminum pistons of similar design were used. The argument, however, goes much further than this, as the engine in which the steel rods are fitted is of 3⅜-in. bore and 5-in. stroke, while that with the aluminum rods is of 4⅛-in. bore and 4¼-in. stroke.

The final result may be computed in terms of cylinder capacity or of piston area. In the former the inertia effects of the steel rod are proportional to $16.100 \div 45 = 0.368$, while for the aluminum rod this "figure of merit" becomes $12.500 \div 55 = 0.228$. Figured on the basis of piston area, the comparison becomes $16.10 \div 8.90 = 1.81$ for the steel rod and $12.50 \div 13.30 = 0.94$ for the aluminum rod.

These figures are sufficiently striking to justify attention, and would be even more remarkable if the engine with steel connecting-rods were designed to reduce the secondary unbalanced forces to the same extent by having the same ratio of connecting-rod length to crank-throw as that in which the aluminum rods are used. This inherent advantage of the short-stroke engine can, however, be thrown in and still leave the argument for the aluminum connecting-rod in its above convincing state.

Summing up, as between two engines, one 3⅜ x 5 in. and the other 4⅛ x 4¼ in., each using aluminum pistons and both of really modern design, the inertia effects that determine the capacity of the engine to resist wear-and-tear of the bearings are reduced by 38 or 48 per cent by the use of aluminum connecting-rods, depending upon whether the cylinder capacity or the piston area is used as a basis for comparison. If it is argued that in the case of the engine with steel rods the wear-and-tear is satisfactory from the user's point of view, the figures then show clearly the possibilities of reducing the size of the bearings and the overall length of the engine, and of the manufacturing economies in respect to the total weight of material required for a given result.

In addition to the above analysis of bearing loading it may be of interest to compare the strength of the aluminum and the steel rods. Considered as a strut, the strength of a connecting-rod is directly proportional to the moment of inertia of its cross-section at the point of maximum stress (which is approximately midway between the ends) and the modulus of elasticity of the material, and inversely proportional to the square of its length. The student will recognize the above as the basis of Euler's formula, which is used as a ready means of comparison for the reason that the relation of the length of automobile connecting-rods to their cross-section does not vary greatly in practice. The moments of inertia of the cross-sections of the connecting-rods are 0.024 and 0.090 for the steel and the aluminum rods respectively.

The relative load-carrying capacity is then $(0.024 \times 30{,}000{,}000) \div (12.125)^2 = 4900$ for the steel connecting-rod and $(0.090 \times 10{,}000{,}000) \div (11.625)^2 = 6650$ for the aluminum connecting-rod, or the relative strengths of the steel and aluminum rods are as 1 to 1.36, the relative piston areas and total explosion pressures being as 1 to 1.5. On this reckoning it will be seen that the aluminum rod is not proportionally so strong as the steel rod. On the other hand, the aluminum rod in question has been subject to most drastic running without a suspicion of failure and is well up to its job.

The truth is that the loading of a connecting-rod is so complex that it is difficult to reduce it to calculation. If engines were run at full throttle continuously at low speeds, the explosion pressure would be the determinant of the design. Just as the speed increases to that at which engines normally run, so do the inertia effects cancel out those of the fluid pressure, while at the very partial throttle required to run an automobile at say 30 m.p.h. the inertia effects completely overwhelm those of the explosion. Further, the compressive stresses set up by the explosion are not nearly so harmful as the alternating stress induced by inertia, so that the above discussion in respect of the explosion pressure is of not much more than academic interest.

In practice it is difficult or impossible to stamp steel rods of sufficiently light section, and subsequent machining is necessary to obtain the best results in respect to strength and lightness. The steel rod in fact suffers from excessive strength and insufficient stiffness. Even if machining is resorted to in order to reduce the weight of the rod, the extent to which this can be done is practically limited to the shank of the rod and the resultant effect is small. For example, on the steel rod in question the reduction of the average section of the rod to 1/16-in. thickness instead of the average 7/64 in. aimed at in forging would reduce the weight by some 3 oz. only, about 6 per cent, an insignificant result compared to that easily attainable by the use of forged aluminum.

Manufacturing Consideration

The general methods of manufacture applied to steel connecting-rods are equally applicable to aluminum. In aluminum rods machining the shank to reduce the weight may be dispensed with, as the consequent reduction of the weight is negligible. Further, there is no necessity for bushing the small end of the rod, although such bushing may be desirable in the case of small-bore engines where the wristpin is necessarily short.

Similarly, with aluminum rods the same babbitted shells may be used as with steel rods, although this practice is regarded as mechanically deficient with either steel or aluminum rods. The use of the babbitted shell necessitates increasing the total weight of the big end to compensate for the additional diameter of the big-end bore required by the shell. More important still, the heat generated by big-end friction *and* that conducted down the rod from the hot region near the piston, has to be dissipated through two oil-films, that between the babbitted shell and the crankpin itself and that between the outside of the babbitted shell and the connecting-rod big-end proper. As the whole object of a bearing is to dissipate readily the heat generated by friction, it is difficult to see why its conductivity should be reduced by 50 per cent.

It may be argued that the advantage of the babbitted shell lies in its capacity for ready replacement. While this may be true, the percentage of bearing failures when babbitted shells are not used is so small as to make it more satisfactory and economical to replace the whole connecting-rod. As in many other instances in automobile design, the provision made for replacement makes the replacement necessary.

On the above grounds the use of direct-babbitted connecting-rods is strongly urged. From the production point of view there is no more difficulty than in babbitting a bronze shell and there is the economy obtained by dispensing with the shell. Successful methods of babbitting aluminum connecting-rods have been developed from the points of view of a complete technical solution of the problem and of rapid economical production. The results of these methods are such that the babbitt in a connecting-rod can be removed only by melting or laborious chipping. There is a definite metallic fusion between the aluminum and the babbitt that it is practically impossible to obtain with steel or bronze.

In the foregoing it has been taken for granted that the virtues of the aluminum piston are generally recognized by engineers, however much they may differ as to whether these virtues are offset by disadvantages. The renaissance of the aluminum piston is beyond doubt, so that it is fair to assume that the aluminum piston has been found to possess a number of advantages. It may not be out of place to state that this is due to the following:

(1) The elimination of piston slap by the use of pistons capable of distorting under high temperature
(2) The reduction of wear by carefully finishing the surface of the cylinder bore and the development of piston alloys of a hardness comparable to that of cast iron

The combination of aluminum piston and connecting-rod allows for a weight reduction of at least 40 per cent in these parts compared with ferrous metals as now employed. The consequences are

(1) That the engine may be speeded up with safety in the ratio of $\sqrt{(100 \div 60)}$, or 30 per cent with the same bearing areas
(2) The bearing areas may be reduced by from 30 to 40 per cent and the engine run at the same speed
(3) Combinations of (1) and (2)

Working along these lines, the author has recently designed an engine with 3¼ x 5-in. cylinders in which the bearing load factor at 3200 r.p.m. does not exceed 14,000 with big-end bearings 1½ in. between the crank webs and 1⅛ in. in net width. With cast-iron pistons and steel rods the corresponding net width of big-end bearing would be about 1⅝ in. The saving in the overall length on a six-cylinder engine is, therefore, some 3 in. in respect of big ends alone, together with further saving of the same amount in the main bearings.

The saving in engine weight arising from this reduction of bearing surface is about 15 per cent, while the available engine speed and torque are all that is required for a car of the medium large type. In other words, a 240-cu. in. engine thus designed with a 4¼ to 1 gear-ratio is capable of doing the work of the average 300-cu. in. engine with a 4 to 1 gear-ratio.

The disposition of weight in an automobile chassis is roughly as given in Table 3.

Of these approximately 25 per cent, notably the frame, wheels and tires, is substantially independent of the chassis weight in that they are dominated in design by body considerations. Treating the engine as a separate unit weighing 25 per cent of the total chassis, we are

TABLE 3—PERCENTAGE DISTRIBUTION OF WEIGHT IN AN AUTOMOBILE CHASSIS

	Per Cent
Engine	25
Frame	10
Wheels and Tires	12
Clutch and Transmission	10
Torque Member, Universal-Joints, etc.	2
Rear Axle	12
Front Axle	3
Radiator and Hood	4
Springs	8
Electric Equipment	6
Steering-Gear	2
Gasoline Tank	1
Miscellaneous	5

left with 50 per cent of the chassis weight varying in some degree with the engine torque. The multiplicity of considerations underlying the design of these parts precludes any definite statement of the extent of this variation, but it is probably safe to say that for equal rigidity of construction the net saving is proportional to the square root of the ratio of the torque under consideration.

Thus the weight of the transmission of a car with a 240-cu. in. engine compared to that of a 300-cu. in. engine, both developing the same brake mean effective pressure, would be as 240/300 = 0.89, indicating an 11-per cent saving in this respect. Summing up, we have a weight reduction of 15 per cent in the engine itself, or some 3.75 per cent of the whole chassis, together with say an 11-per cent reduction in the weight of 50 per cent of the chassis, or 5.5 per cent of the whole; in all some 9 per cent arising indirectly from the use of aluminum connecting-rods and pistons. In the case of a chassis weighing 2400 lb. the net result is, on the above reasoning, a saving of 216 lb. The weight of the cast-iron piston and steel connecting-rod in a six-cylinder 300-cu. in. engine would approximate 36 lb., while their aluminum counterparts would weigh some 21 lb. Thus a saving of 15 lb. in pistons and rods results in an overall saving of about 14 times this amount.

To forestall criticism, no one is more aware than the author is of the vagueness of this estimate, due to the vast number of factors that enter into the problem. On the other hand, there is no doubt of the truth of the general proposition as to the enormous benefits to be derived by reducing the weight of reciprocating parts by the use of light-weight alloys. The limitations of this paper unfortunately preclude any discussion of the potentialities that lie in the application of these materials throughout the chassis. The fact that extended experience has shown them structurally suitable for the hardest worked parts of the whole chassis indicates the confidence with which they may be applied elsewhere.

THE DISCUSSION

David Fergusson:—Mr. Pomeroy's paper is so convincing that it is somewhat difficult to give good reasons for not following his advice. However, there are a few points that I would like to call attention to. In the table giving the specific strength of various automobile materials, the figures of most consequence are those of the elastic-limits of these materials, rather than those of the ultimate tensile-strengths. If this be conceded, it changes the position of the forged aluminum materially. I believe I am right in stating that the elastic-limit of forged aluminum is about 25,000 lb. per sq. in. Its specific strength in relation to its elastic-limit is therefore 133. The elastic-limit of 0.35-per cent carbon-steel heat-treated to give 105,000-lb. tensile-strength will be about 80,000 lb. per sq. in., giving a specific strength of 163, or nearly 25 per cent greater than that of forged aluminum. The elastic-limit of 3-per cent nickel steel heat-treated to give a tensile-strength of 170,000 lb. per sq. in. will be about 130,000 lb. per sq. in., giving a specific strength of 263 or double that of forged aluminum.

A doubtful point in connection with this comparatively new material is its life or endurance. Mr. Pomeroy has had personal experience with this in service, which is the real test, yet a test I had made about 3 years ago in a Stanton fatigue-testing machine gave very poor results, the specimen failing after 1092 blows of a weight falling 1 in. Common screw-stock of about 0.20-per cent carbon-steel stood 6000 blows. Forged aluminum, no doubt, has been improved since this test was conducted.

The hardness of forged aluminum is I believe only about 100 Brinell, compared with 230 for heat-treated 0.40-per cent carbon-steel. If this is so, is there not trouble due to the metal peening-out? This would result in the small end of the connecting-rod enlarging on the piston-pin if no bearing bushing were used, or in the bushing becoming loose, if one were used. Is there any trouble due to the big-end bolt-heads peening their way into the softer aluminum? Is it necessary to use a large steel washer between the connecting-rod cap and the nuts on the big-end bolts?

Is there not some trouble from the greater expansion of the small and big ends of the aluminum rod, giving these a greater clearance than desirable and so causing a knock? This is one of the troubles I have had with aluminum pistons in which the piston-pin floated in the piston. Unless these were assembled very tight when cold, there would be a knock when the piston warmed-up.

Is not the cost of the aluminum forging somewhat excessive? When I looked into this matter three years ago I found that the cost was so great that it would pay to use steel and machine the rod all over, as the saving in weight would then be very little, the only large saving being in the big end, as I considered, perhaps wrongly, that the cross-section of the rod should be three times that of the 0.35-per cent carbon-steel, which is the material I have used on all medium-speed engines, the stiffness being as great as that of the higher-priced alloy-steels. The elastic-limit of this steel is three times that of the aluminum and the modulus of elasticity is about three times that of aluminum, while the specific weight of the aluminum is less than one-third that of steel. In making the comparison, I, of course, figured on a bronze bushing in both the large and small ends of the aluminum connecting-rod.

The saving in the length of Mr. Pomeroy's engine, due to the use of aluminum pistons and connecting-rods, is certainly very interesting. However, I have found that the length of a six-cylinder engine is largely controlled by the diameter of the cylinder bore; the wall thickness; and the space for water between the cylinder barrels, which I consider a necessity. The last named cannot be less than ¼ in. and should be more to satisfy foundry requirements. The diameter of the crankshaft must be so large to avoid excessive torsional vibration that the length of the bearings that the above conditions admit of are usually ample for all medium-speed engines.

I would like to hear more of the type of aluminum piston Mr. Pomeroy has had most success with, including how he avoids piston slap when cold and scoring when hot. I believe that there is a great future for the use of forged aluminum in automobile construction. Mr. Pomeroy has done much to show the way.

L. H. POMEROY:—Mr. Fergusson, with characteristic thoroughness, puts up the contra side of the aluminum versus steel argument. I cannot, however, let his figures on specific strength in terms of elastic-limit pass without comment. In the first place, the elastic-limit of any material is most difficult to ascertain. In fact, the only physical properties that can be determined with anything like accuracy are the ultimate stress and the elongation. It is upon the former of these that the vast bulk of safety factors are based. Without going into this very vexed question, let us see what happens if, instead of taking the elastic-limit as a basis for determining the specific strength, we take another quantity that is related thereto but more easily measured, namely the yield-point.

It is well known that by suitable heat-treatment carbon and alloy-steels can be made to give a very high ratio of yield-point to ultimate tensile-strength at the expense of elongation, but such high ratio and consequent brittleness by no means make the material more suitable for practical purposes. In other words, experience shows that elongation is a necessary characteristic of most materials of construction and that it must be obtained even at the sacrifice of a high yield-point. This in itself to a large extent invalidates Mr. Fergusson's basis of comparison unless such basis predicates the same elongation as may be obtained with wrought aluminum. The yield-point of a good wrought aluminum alloy with an elongation of 20 per cent is approximately 35,000 lb. per sq. in. with an ultimate stress of 60,000 lb. The specific strength in terms of the yield-point becomes therefore the yield point divided by the weight per cubic foot, or 35,000 ÷ 189 = 185. The physical characteristics of S.A.E. No. 1035 steel, having a carbon-content of 0.35 per cent, as given in the S.A.E. HANDBOOK show that when this material is treated to give a 20-per cent elongation it has an ultimate stress of 95,500 lb. per sq. in. and a yield-point of 64,500 lb., the specific strength in terms of the yield-point being 64,500 ÷ 490 = 132.

Similarly, S.A.E. No. 2330 steel, which is a 3-per cent nickel 0.30-per cent carbon-steel susceptible of being heat-treated to give 170,000-lb. ultimate stress, possesses only an elongation of some 11.5 per cent in this condition. When treated to possess an elongation of 20 per cent the ultimate stress becomes 104,000 lb. per sq. in. and the yield-point 77,000 lb., the specific strength in terms of the yield-point being 77,000 ÷ 490 = 157. These figures show then that the specific strengths of wrought aluminum, 0.35-per cent carbon-steel and 3-per cent nickel 0.30-per cent carbon-steel, all with an elongation of 20 per cent, are 185, 132 and 157 respectively, instead of having values of 133, 163 and 263 as given by Mr. Fergusson.

With respect to endurance or resistance to repeated stress, one cannot of course expect all the poor results to be found with steel only. I would only remark that there are few tests that seem to have provoked more argument as to their value than fatigue tests in general. In actual practice automobile engine connecting-rods can be made of aluminum with from 55 to 60 per cent of the weight of a steel connecting-rod, including bolts, babbitt, etc., and have stood up under the most strenuous conditions. In the aluminum car I have designed we run the engine at speeds of over 3000 r.p.m. with impunity and in a collective mileage on four cars of nearly 80,000 miles and on one car of some 40,000 miles there has not been a symptom of failure. Forty thousand miles with a gear-ratio of 4.25 to 1 is approximately 1780 hr. running at 1000 r.p.m., or 1780 × 60 × 1000 = 107,000,000 revolutions, or 214,000,000 strokes. Allowing three stress reversals per cycle, this becomes over 150,000,000 stress reversals.

Some fatigue experiments made in a fatigue-testing machine consisting of a motor-driven crank and weighted cross-head may be of interest. At 1500 r.p.m. the wrought aluminum rod gave a life of 353 hr. when the crosshead itself broke, whereas the steel rod failed in one case after 25 hr. and in another after 45 hr. of running under identical conditions. The steel rod was of smaller section but about double the weight of the aluminum rod.

So far as peening-out is concerned no trouble has been experienced, nor should there be any if sufficient metal is used around the small-end bearing. With steel rods, of course, a bronze bushing that does not peen-out is universally used, so that the comparison between the Brinell hardness of wrought aluminum and of 0.40-per cent carbon-steel is hardly appropriate.

Steel washers are certainly desirable and in fact necessary to avoid trouble arising from the nut of the connecting-rod bolt or lock-washer cutting the aluminum when being tightened.

The difference in expansion of aluminum in contact with steel is about 0.001 in. per in. per 100 deg. fahr. Given a good initial fitting of the wristpin and adequate lubrication, no trouble is experienced, although doubtless these conditions are more important than with a steel rod unless the aluminum rod is bushed, in which case the requirements are identical. It is not the least of the advantages of the aluminum rod that the usual bronze bushing in the small end may be eliminated.

The cost of aluminum rods nowadays competes with an unmachined steel rod if the big end of the aluminum rod is direct-babbitted, and is much less than that of a steel rod machined all over.

I think that Mr. Fergusson's remarks on overall engine length apply primarily to a T-head engine. Admitting his premises, there is no reason why the bore cannot be reduced, which shortens the engine and reduces weight to a greater extent than it is increased by the longer stroke required for a given cylinder capacity.

The aluminum pistons to which Mr. Fergusson refers are of the split-skirt type made by the United States Aluminum Co. These pistons are characterized by the nature of the piston skirt, which is split to allow for expansion, and by the section of the skirt which is designed to allow deformation to take place without causing the portion of the skirt that is in contact with the cylinder to go out-of-round.

A further important feature is the discovery of a process for making these pistons up to 150 Brinell hardness, which has a marked effect upon their resistance to wear. In the engine previously mentioned these pistons are put up with 0.004-in. clearance for a 4⅛-in. bore and no trouble has been experienced from seizure or slap.

The successful aluminum piston, like the successful aluminum connecting-rod, cannot, however, be designed blindly. There is a definite technique of construction that has to be observed and those who imagine they can substitute aluminum for steel or cast iron without modification had better not consider the matter further.

MR. FERGUSSON:—Have you any trouble with scoring of the cylinder?

MR. POMEROY:—Not that I know of.

E. H. SHERBONDY:—Is the constant you give for loads of 16,000 Ricardo's or your own? And how was it arrived at?

MR. POMEROY:—The constant was arrived at empirically by Mr. Ricardo, and is in conformity with my own experience. It is the average loading or pressure taken on the crankpin bearing.

OTTO M. BURKHARDT:—I notice that there was rather a sharp line of demarkation between Mr. Fergusson's and Mr. Pomeroy's figures. They represent, so to speak, two different schools of design. Mr. Pomeroy is basing his calculations entirely on the tensile-strength and Mr. Fergusson is basing his calculations on the elastic-limit. As Mr. Pomeroy says, the elastic-limit is rather an indefinite figure, whereas the tensile-strength is well defined and can be obtained, even with crude testing-machines. All fatigue tests that I know of have invariably been formulated on the basis of tensile-strength. During some very interesting tests made at the University of Illinois, it was found that the formulation of the results can be made to better advantage on the basis of the Brinell hardness. There is a fairly definite relation between the Brinell hardness and the tensile-strength of heat-treated steel. This relation has been established by John Miller, the metallurgist of the Pierce-Arrow Motor Car Co., and is represented approximately by

$$\text{Tensile-Strength} = 500 \times \text{Brinell Hardness}$$

No similar relation can be given for the elastic-limit but a very similar relation can be given for the yield-point. It is, according to Mr. Miller,

$$\text{Yield-Point} = 550 \times (\text{Brinell Hardness} - 75)$$

From this it follows that, when factors of safety are based on the yield-point, a happy compromise can be obtained between the two schools here represented. Mr. Pomeroy has chosen the tensile-strength for the simple reason that there is no well-defined elastic-limit in the case of aluminum. Mr. Fergusson has chosen the elastic-limit because this can, through patient research, be found for steels, and Mr. Fergusson, I take it, is a sponsor for the use of steel. In the calculations that I have carried out I have found it most satisfactory to base the factors of safety on the yield-point where infrequent shocks are under consideration. Where fatigue is under consideration, it is advantageous to deal with the tensile-strength. I have analyzed somewhat further the relative merits of the metals here under consideration. If we denote the tensile-strength of ferrous metals by T_f and the tensile-strength of aluminum by T_a, the ratio between the two is

$$T_f/T_a = K$$

This relation indicates that ferrous metals are K times as strong as aluminum, although K may well be smaller than unity.

If we further take into consideration the specific gravity of the two metals, we have another factor that rather expresses the inverse of the previous factor, namely the density of aluminum relative to ferrous metals. This factor is rather constant and may easily be taken as

$$C = 2.6/7.7 = 0.338$$

In case the structural part be subject to either pure tension or compression, it is obvious that the product of the two factors, namely K times C, would represent the necessary weight to be employed for an aluminum or steel rod or bar respectively in order that in either case the same factor of safety for a steady load may be obtained. I have made special mention of a steady load, because the factor of safety would be an altogether different one in the case of a rapidly fluctuating or reversing load, as in such a case fatigue would have to be considered and aluminum is considerably inferior to steel so far as fatigue is concerned. The product $K \times C$ is a direct function of K only and it is an easy form to represent graphically. For instance, if we plot different values of K as abscissas, and as ordinates we plot the product $K \times C$, as in Fig. 1, we can determine at a glance from the ordinates the weight of an aluminum part corresponding to an equally strong ferrous metal part.

In case of bending, we have a slightly different problem, as we then have to take into consideration the sectional modulus. We may agree on a section of let us say a width equal to three-eighths of its height and inasmuch as the sectional modulus is determined by the width and the square of the height divided by 6, we have in case of ferrous metals

$$3/8\, h^3 \div 6 \text{ or } 3\, h^3 \div 48$$

Denoting with h_a the corresponding dimension for an aluminum section, it is obvious that the section should be K times as strong as the ferrous metal section and consequently have the following relation:

$$h_a^3 \div 48\, K = h^3 \div 48$$

From this it follows that

$$h_a = h \sqrt[3]{K}$$

Inasmuch as the weight is proportional to the area of the section, we would have a relation between the weight of a ferrous metal lever and that of an aluminum lever,

$$(3\,h_a^2/8) \div (3\,h^2/8) = h_a^2/h^2$$

Substituting for h_a, we obtain

$$(h^2\,\sqrt[3]{K^2}) \div h^2 = \sqrt[3]{K^2}$$

If we multiply this weight ratio by our previously obtained constant, $C = 0.338$, we have a direct relation between the weight of an aluminum lever and that of a ferrous metal lever, both being designed to give the same factor of safety.

In cases where the deflection is of the greatest importance we must bear in mind that the moment of inertia is the determining factor, and this is determined by the fourth power of the sectional dimensions. For instance, the moment of inertia of a section similar to that previously considered for bending would be expressed by

$$3/8\,h^4 \div 12 = 3\,h^4 \div 96$$

Inasmuch as the modulus of elasticity of steel is approximately three times as large as that of aluminum,

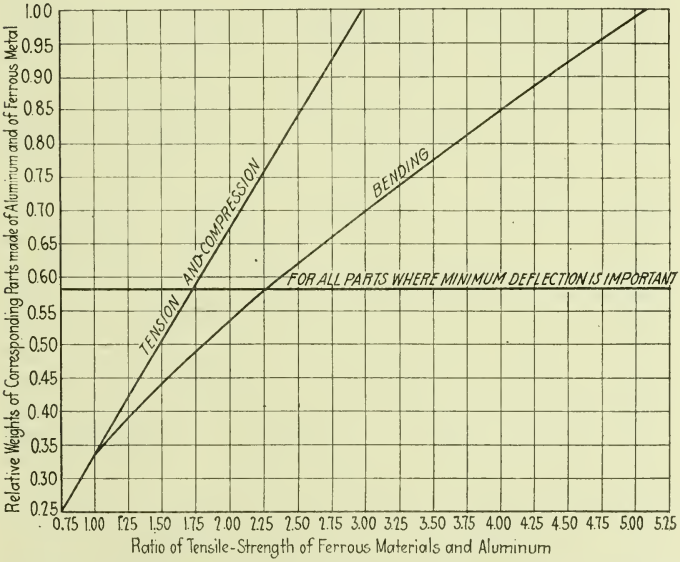

FIG. 1—CHART SHOWING THE RELATION BETWEEN THE RELATIVE WEIGHT OF CORRESPONDING PARTS MADE OF ALUMINUM AND FERROUS METALS AND THE RATIO OF TENSILE-STRENGTH OF FERROUS METALS AND ALUMINUM

it is obvious that the aluminum section should be such that the moment of inertia is three times as large as that pertaining to the steel section. This insures equal rigidity and may be mathematically expressed by

$$(3\,h^4 \div 96) \times 3 = 3\,h_a^4 \div 96$$

From this it follows that

$$h_a = h\,\sqrt[4]{3} = 1.3161\,h$$

For a comparison of weights, we have to consider again the areas and similarly as before we have

$$3/8\,h_a^2 \div 3/8\,h^2$$

After substituting for h_a, we obtain

$$[3/8 \times (1.3161)^2 \times h^2] \div 3/8\,h^2 = 1.732$$

In other words, 73 per cent more area is required for aluminum than for steel in order that both sections may be of equal rigidity.

If we now multiply this constant factor by our factor C, we obtain

$$1.732 \times 0.338 = 0.585$$

Or in a case where deflection is to be held to a minimum, an aluminum part of only 58½ per cent the weight of a steel part can be substituted with equal satisfaction.

In conclusion, I would say that with steels we have reached some sort of an obstacle between what can be had out of the steel in the laboratory and what the factory can handle successfully. We can heat-treat alloy-steels easily to give a yield-point of over 200,000 lb. per sq. in. However, it would be utterly impossible with our existing cutting-tools to handle a steel thus treated successfully in the factory. It is, therefore, necessary to machine steel parts while yet annealed and heat-treat them after machining. This, as we well know, involves scaling and distortion and requires grinding after heat-treating. The only alternative is to sacrifice the best that can be had from steel and be satisfied with the heat-treatment giving a yield-point of only half of what the steel is perhaps capable of, and steel so heat-treated can be handled successfully in the factory. No such limitation is encountered in the use of aluminum. In fact, we are far from getting aluminum hard enough. It is necessary to look forward to a new development of cutting tools to give us greater speeds so as to utilize thoroughly this outstanding property of aluminum that we know is easy cutting.

MR. POMEROY:—Mr. Burkhardt's contribution is an important supplement to my paper. I may say that in general the substitution of aluminum for steel is most easily achieved when the steel part is bounded by the atmosphere. In the case of a crankshaft, for instance, although it might be possible to make this of aluminum and save weight in itself, the necessary increase in the diameter to obtain strength and stiffness and the further increase in the weight of bearings, due to the increased size, would practically balance the initial weight-saving on the crankshaft itself. The case of a connecting-rod is, however, very different and there are usually no pronounced limitations in the space available for the increased section required. The case is similar with an automobile frame. The car to which I have referred has a cast aluminum frame that has stood up perfectly under the most arduous conditions of road use. Its weight is about 60 per cent of that of a corresponding steel frame. In this particular instance the strength is conferred by the dimensions, while the material is of relatively low tensile-strength.

E. O. SPILLMAN:—We have been experimenting recently with a piston with slots, the piston having the ordinary clearances. Some of the test pistons developed piston slap. I have not taken them down to find out what the trouble is, but I think these pistons have collapsed on the off-pressure side. Should we increase the weight of this piston or increase the Brinell hardness? Does Mr. Pomeroy use aluminum shims on the big end?

MR. POMEROY:—If the piston has a slap as you describe, I think that it has collapsed. If this is the case, more metal is needed. We certainly do not recommend shims in the large end of connecting-rods.

MR. SHERBONDY:—Mr. Pomeroy compared the piston and connecting-rod weights in the Essex engine with his own on a basis of cubic-inch capacity. I believe that these should be based on the horsepower output at any given speed. What does Mr. Pomeroy consider a fair stress for connecting-rods? And what does he consider a safe deflection for them?

MR. POMEROY:—I agree with Mr. Sherbondy that the basis for comparison of the weights of connecting-rods in various engines should be the horsepower developed at any given speed, since this is in terms of cylinder capacity if the brake mean effective pressure is the same in the two engines. In the case in question, the brake mean effective pressure of the Essex engine is about 10 per cent higher than that of my own engine and this correction though small should be allowed for.

The safe stress for an aluminum connecting-rod is rather difficult to state as the loading of a connecting-rod at high speeds is very complex. Using more or less accepted methods of calculation, I try to keep the combined stress in the shank of the rod down to about 5000 or 6000 lb. per sq. in. at 3200 r.p.m. This can usually be done if the section of the steel rod is increased by 20 per cent. Each case, however, demands individual consideration. Many steel rods in use are stiffer than they need be from forging considerations. In other words, forged connecting-rods would be considerably improved if the shank section were reduced by machining.

A MEMBER:—What of the possibility of using metallic magnesium for the same purpose as aluminum?

MR. POMEROY:—The use of magnesium is now being developed for a considerable number of automotive parts. Its application to connecting-rods is, however, a matter upon which nothing can be said at the moment.

A. J. FITZGIBBONS:—Has a successful universal-joint been made of aluminum?

MR. POMEROY:—In the case of a universal-joint the difficulty which arises is that of fixing the aluminum forging to the shafts themselves. There is no real reason why this cannot be done but so far the circumstances have not arisen to cause it to be done.

MR. BURKHARDT:—How about the ring type of universal-joint?

MR. POMEROY:—I do not see why aluminum could not be used for that.

MR. SHERBONDY:—Another point is the question of expansion, where the connecting-rods and the diameters of the pistons are small. Here we have to deal with 3 or 4-in. diameters and run at from 0 to 160 deg. fahr. temperature, so that the change in size becomes a very serious factor. In some cases it may cause failure. In fitting pistons as tightly as Mr. Pomeroy recommends, the only way to fit them is to heat them before putting them into the cylinders.

MR. POMEROY:—The expansion of aluminum is twice that of steel. For a 2-in. diameter shaft, a 100-deg. temperature-difference between the steel and the aluminum would mean a difference in the diameter of 0.001 in. and it is difficult for me to believe that this would make any great difference.

AUTOMOTIVE BRAKE AND CLUTCH PRACTICE HERE AND ABROAD[1]

By H G Farwell[2]

The author describes the major features of brake and clutch practice that he observed in 1920 while traveling in England, Belgium, Italy and France, comparing them briefly with American practice of the same period. He analyzes the types of brake and clutch used on 165 cars exhibited at the London automobile show of that year, giving the percentage of the different types in evidence.

Numerous illustrations that are described and commented upon in greater or less detail appear in the paper and in the discussion which followed it, these being inclusive of most of the best-known types of brake and clutch in use in the United States and in Europe.

Before recounting the major features of European brake and clutch practice that I observed during several months of travel in England, Belgium, Italy and France in 1920, I will outline the general practice in the United States as it existed at that time. The ordinary type of external brake was in general use for service work; in Europe it is known as the foot-brake. What we call the emergency brake is known as the hand-brake in Europe because it is operated by the hand-lever; it is nearly always cam-operated in this Country. There were also some cases in which one brake, sometimes the service and sometimes the hand-brake, was used on the transmission; cases in which both sets of brakes were internal, on the rear wheel-drums; cases where these were in concentric drums; and, in some instances, the two brakes were placed side-by-side. Regarding clutch practice in the United States, we had seen a gradual reduction in the number of cone clutches used, until comparatively few cars are so equipped in proportion to the number of cars built. We have seen a gradual increase in the use of the disc clutch; the single-driven-plate and multiple-disc types being used in high-grade cars. One striking feature is the size of brake-bands. In this Country the diameter of the rear-wheel brakes usually is from 35 to 45 per cent of that of the rear wheels. It is not unusual to find 12, 14 and 18-in. drums on cars weighing from 2100 to 5500 lb.

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[1] Metropolitan Section paper.

[2] M.S.A.E., Chief engineer, Raybestos Co., Bridgeport, Conn.

ENGLAND

In England the use of the internal brake was fairly general on all classes of cars and, with comparatively few exceptions, the use of the transmission brake was rather more marked there than in this Country. In general, very little criticism was found on the operation of these brakes, but there is one exception in the case of the Daimler car which, until recently, has always used an external service-brake on the rear-wheel drums. I understand that the company is planning now to change that to the conventional internal expanding type. On several of the post-war cars weighing 1600 lb. and even less, they were using external band brakes. The objection made in England to the use of the external brakes is that they always drag. Their engines are very much smaller, for fuel-economy reasons; any dragging of brakes uses up a greater proportion of available engine power than is the case with cars in the United States.

The usual English criticism of American brakes is that they are intolerable, although I found about as many unsatisfactory brakes in England as here, on from one to five makes of car, depending upon the locality in which they are used. The brakes on city cars generally are maintained better than the average brake used in the country districts. Probably more attention has been given in England to brake design and the construction of internal brakes than here; in fact, that is true throughout Europe. The main objection to the usual internal brake is that wear is confined to a very much smaller area than is the case with external brakes. To compensate for this, the English designer arranges the adjustment feature so that brake adjustment can be made much more easily, often without getting out and going under the car. In the English models that have been produced within the last 2 years, the ease of brake adjustment has received great attention. In some cases large handwheels are placed so that one can adjust the angle of the cam with but little more difficulty than that of getting out of the car.

I had been led to believe that metal brakes were always noisy, but I found this to be an exaggeration. Except for the metallic grind, these internal brakes were only slightly more noisy generally than the American conventional type of external brake. I attribute this to better or closer fitting and to the elimination of points where vibration may occur. One finds noisy brakes in England, the same as here; but, in general, they are not much worse there. Brake-drums are very much smaller in England. To cite one case, an American car weighing approximately 2100 lb. was equipped with 31 x 4-in. tires and 14-in. brake-drums. The English car, having exactly the same size of engine, weighed 2800 lb., had 30 x 1½-in. tires and the diameter of the brake-drums was 10 in. After looking over the field, I am sure that the tendency in this direction is to increase the diameter of the brake-drums. Probably tire sizes will always be kept smaller, for the sake of economy and first cost, but the brake-drum diameters are certainly growing.

The English clutch situation was interesting in that it showed so many cone clutches on cars of the higher grades. The single-plate clutch also is used, but not to

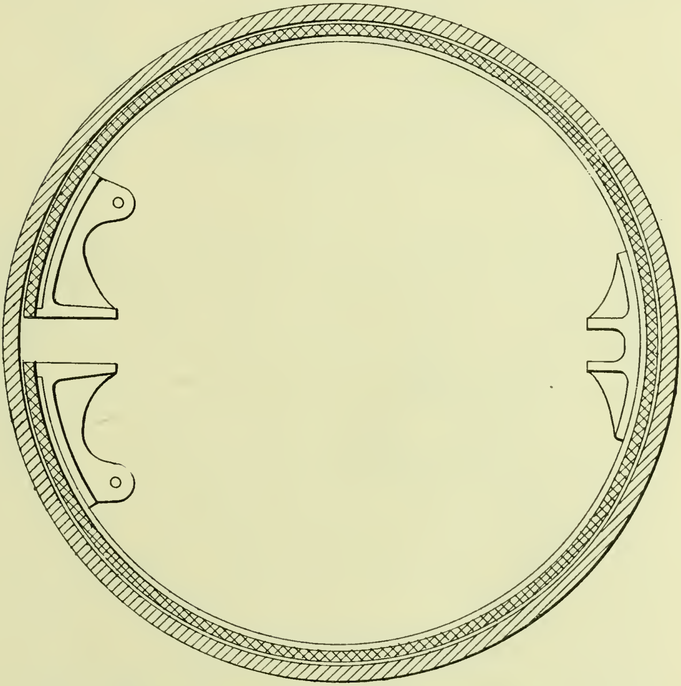

FIG. 1—CONTINUOUS BAND TYPE OF BRAKE

the same extent as here. There is a noticeable tendency to consider the use of the plate clutch, instead of the cone type. It will be surprising to see the great number of plate clutches that will be used in the years to come. There is less need for large-capacity clutches in England than there is here. That has a bearing on the use of the multiple-disc clutch. It is interesting to note some of the tendencies at the automobile show held in London in November, 1920. I have not checked these figures with those recently published; they are simply the result of my personal observation.

Seven nations were represented by 165 exhibits, Switzerland had 1; Holland, 1; Belgium, 3; Italy, 7; France, 28; the United States, 30; and England, 95. Of these 165 exhibits 49, or practically 30 per cent, were equipped with clutches of the single-plate type. There were 28, or 17 per cent, with clutches of the multiple-disc type. Only 4, or 2.5 per cent, had friction drives; that is, having the disc bearing on the ring, about 1 to

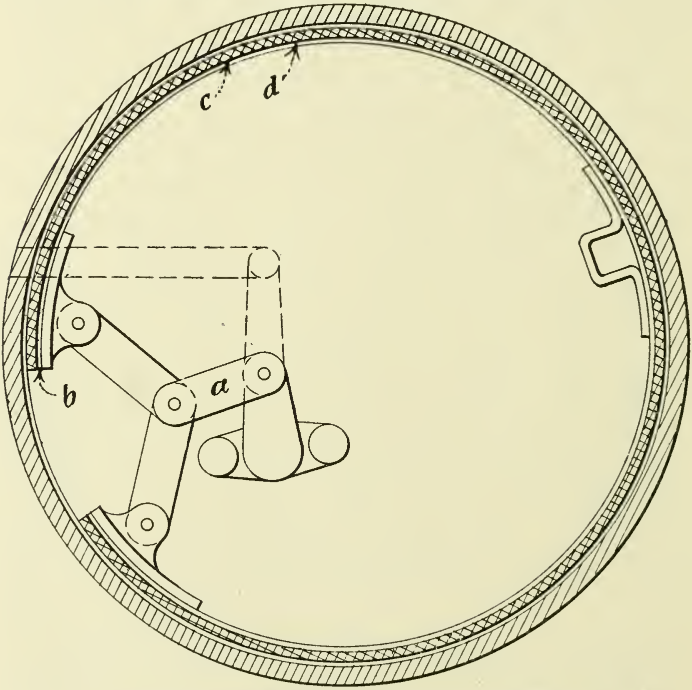

Fig. 2—Another Internal Brake That Is Operated by Two Links

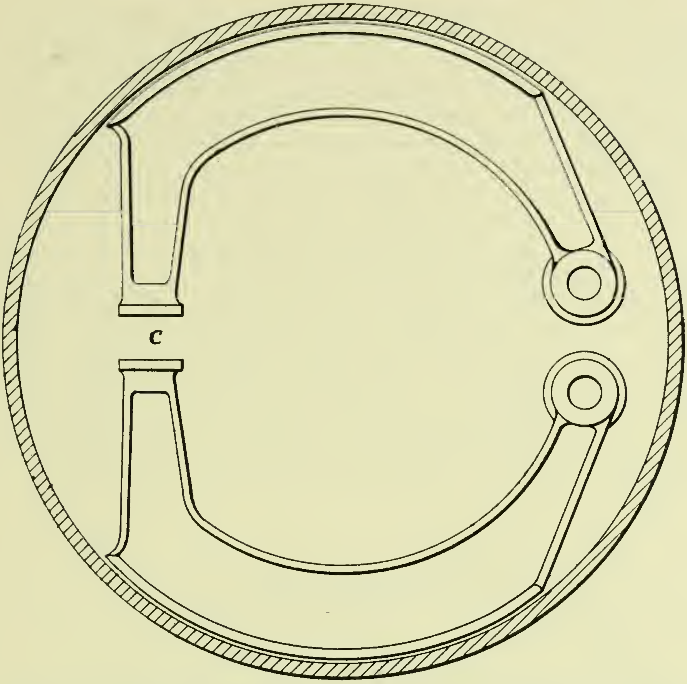

FIG. 3—A EUROPEAN INTERNAL BRAKE WITH TWO OPERATING PIVOTS

1½ in. wide. This leaves 83 cars, or just over 50 per cent, that were equipped with cone clutches and with linings and leather, cotton or asbestos. Of the 95 English cars 56, or 59 per cent, had cone clutches. There were 11, or 12 per cent, with clutches of the multiple-disc pattern and 24, or 25 per cent, had clutches of the single-plate variety. The only friction-drive clutches of the 165 exhibits were of English manufacture.

Of the French cars 61 per cent had cone-type clutches, 18 per cent were of the plate and 18 per cent of the disc type. The only band clutch was of French make. Of the 30 American cars, 50 per cent were equipped with single-plate, 27 per cent with multiple-disc and 23 per cent with cone-type clutches. With one or two exceptions, all the 30 American cars used the external brake for the service or foot-brake. Perhaps four or five used a transmission brake but, in general, the usual external and internal sets were used. With two or three exceptions, the English cars used internal brakes; most of them had trans-

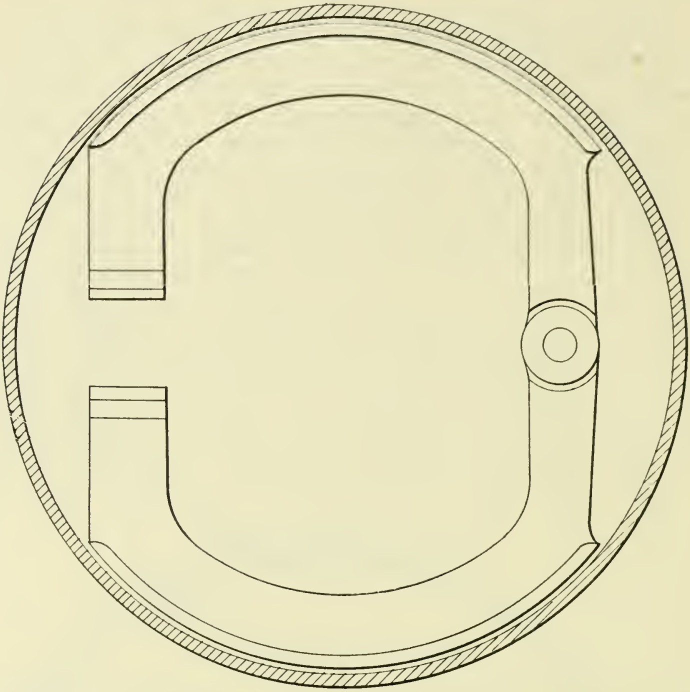

FIG. 4—IN THIS EUROPEAN BRAKE ONLY ONE PIVOT IS REQUIRED FOR OPERATION

mission brakes also. The Continental cars stand about the same as the English with respect to brakes, but we often find the external brake used on the transmission.

CONTINENTAL COUNTRIES

The brake situation in Belgium is very similar to that in England; the internal expanding type is used almost universally and generally the brakes are metal-lined. There are cases where an asbestos liner is used and usually the brakes are fitted so that it can be used if so desired. Here also we find a considerable interest in the disc type of clutch. Of some eight or nine firms, there is a greater proportion that build an engine that rates over 15 hp. than in England, although the actual number is really less. While those who use a cone clutch declare that they have no trouble with it, they are still a bit more than willing to try a clutch of the plate or multiple-disc type.

In Belgium we find one of the strong proponents of brakes on four wheels. The operating practice of one firm varies in a very interesting way from some others, as will be illustrated later. We find the internal brakes on the front-wheel drums usually operated by cams at the tops of the drums. This operating cam is on one end of the shaft that carries the lever and is fastened to the side-member of the frame. This shaft is furnished also with a flexible joint which allows for the steering movement of the wheels and carries the cam ring of the shaft. In the Belgian construction mentioned the brake is operated from the bottom and, by placing a specially shaped cam in the actual line of the steering-knuckle, carrying the operating shaft round with the wheel during the steering movement is avoided. I think there are only eight or nine cars built in Belgium, so that the field and the differences of practice are somewhat limited.

In Italy we find the internal brake and a rather larger

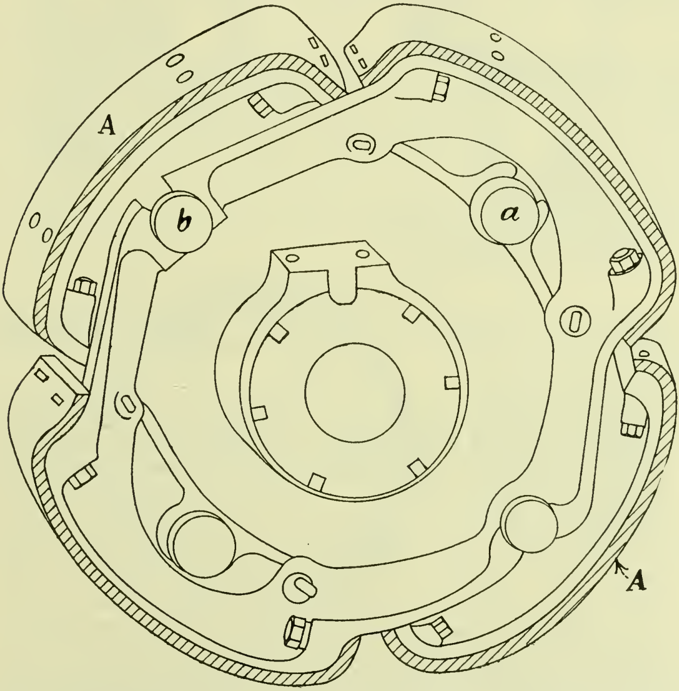

Fig. 5—A Double Internal Brake in Which Not More Than One-Quarter of the Drum Circumference Is Covered

proportional use of the transmission brake than in some of the other countries. The center of the industry in Italy is in the north, around Milan and Turin. Perhaps on account of the physical characteristics of the country, their engines are larger on the whole and their cars heavier and more rugged. Brakes are considerably more closely watched there than in England. Metal liners are generally in use. Brake-drums follow the other practice closely. We find brakes on four wheels in two or three cases. This feature is often made optional with the purchaser, at a price. The cone clutch is surely going out and more and more of the multiple-disc type are being used. An interesting feature of the multiple-disc type of clutch in Italy is that the diameters of the plates are considerably greater than those of the American clutches of the same type. The general practice in multiple-disc work here runs between 7 and 9 in., while the multiple-disc clutches of the Italian cars will run between 8½ and 10 in. Most of the multiple-disc

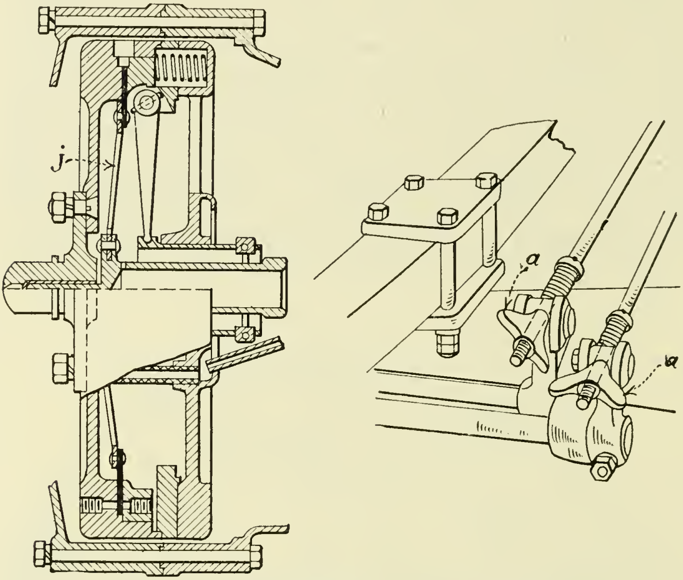

Fig. 6—Two Examples of European Design

The View at the Left Shows a Clutch Having the Facing on Both Sides and the Springs at the Periphery While the Mechanism Employed for Operating the Brakes on a Belgian Car Is Illustrated at the Right

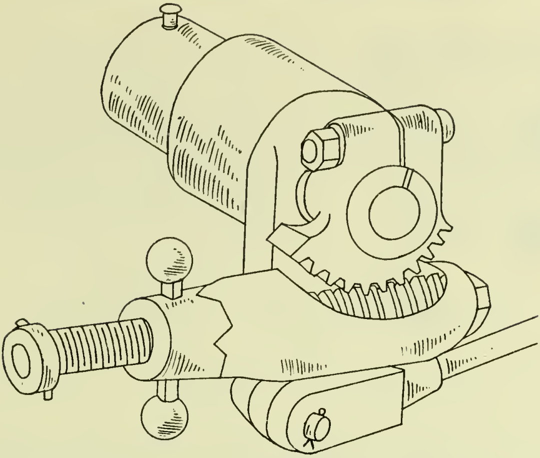

Fig. 7—A Form of Brake Adjustment in Which the Angularity of the Camshaft Is Changed

clutches have liners of some kind. In one case cotton was used.

France is one of the most interesting centers of the automotive industry as regards new applications of ideas. We find the use of the internal brake on the rear wheels almost universal there; and the drums are being increased in size considerably. There is a peculiar intermixture of the use of internal and external brakes; internal brakes on the rear wheels and an internal brake on the transmission, or an external brake on the trans-

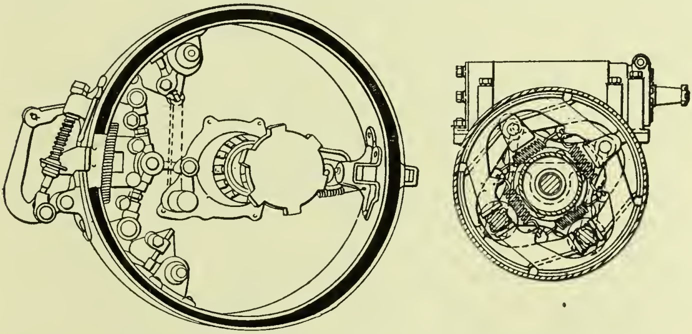

Fig. 8—Two Forms of American Truck Brake

At the Left Is a Simple Type of Toggle Brake and in the Brake at the Right a Certain Amount of Movement of the Cam Lever Is Necessary before the Lost Motion between the Shoe and the Drum Is Actually Taken Up

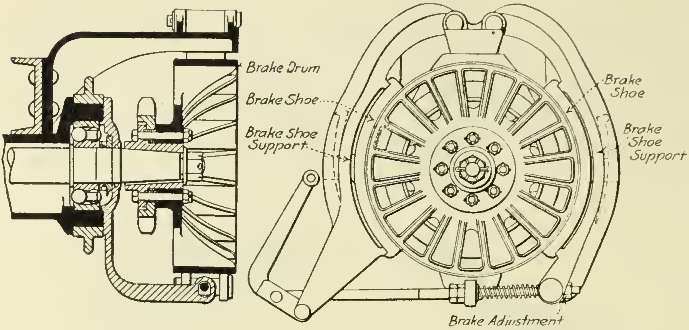

FIG. 9—AN EARLY FORM OF BRAKE IN WHICH AN EFFORT WAS MADE TO AIR-COOL THE DRUM

mission and internal brakes on the rear wheels. Some of the larger cars have brakes on four wheels. Usually they are listed as "extra." The matter of applying brakes and their operating connections to the steering-knuckles offers some peculiar difficulties. The one interesting application is on the Hispano. The brakes on the four wheels are operated by an auxiliary clutch that is brought into action by the foot clutch rather than by spring action. This auxiliary clutch operates on a shaft that is driven by a worm and gear from the transmission shaft. Considerably greater attention is given to the equalization of brakes, as a rule. Anyone who has used or driven a car equipped with brakes on its four wheels cannot help but be impressed by the ease with which a car is controlled and the quickness with which it can be brought to a sudden stop. By "sudden" I mean a stop that will bump one's head against the windshield.

France still uses many cone clutches. Several multiple-disc and some single-plate types are used. One very interesting clutch is of the single-plate type; the clutch facing is required to do double duty, both sides being used. The facing is fastened to the metal plate that fits snugly inside the center hole of the facing. Another clutch that shows some ingenuity is one in which the shaft on which the driven member moves has been eliminated. The requisite motion for release is obtained by the buckling or dishing of the metal driven plate.

European practice is so different from our own that we cannot criticise; it is based on economy, speed and comfort. The designers in France, England and Europe in general have just as much reason for their designs as we have for our own. French engineers tell me that French automobile body lines today are following those of American cars more than ever before. They also are following our practice of unit powerplant construction in some cases, but we find that the physical characteristics of the country and the gasoline price have a bearing upon European design and, when we criticise, we should take those factors into consideration.

BRAKES

Fig. 1 shows a continuous-band brake. One of the great difficulties with this type is the excessive wear on the cam ends. The back usually becomes clogged with mud in spite of the fact that it is an internal brake, and it often refuses to operate. Fig. 2 shows another brake operated with two links, and this practice is subject to the same trouble. The pressure at *a*, instead of being entirely tangent at *b*, is nearly radial. The wear is on the facing at *c* and there is practically no wear at *d*.

Fig. 3 shows one of the internal brakes used in Europe, and unusual in American practice; it differs in having the two pivots at *a* and *b*, with the usual cam-operating mechanism at *c* except that it is brought back from the circumference of the drum. Fig. 4 shows an internal brake of the same general type, except that it has only one pivot at the back. In two or three cases the brakes are operated by a wedge instead of a cam. Fig. 5 shows a double internal brake having only about one-quarter

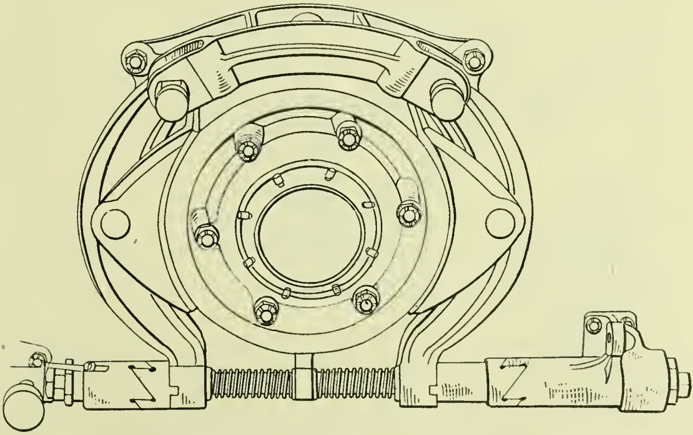

FIG. 10—THE PIERCE-ARROW BRAKE USING A RADICAL CAM

or less of the circumference of the drum covered. One set of brakes, *A*, is operated from the cam *a* and the other set from the cam *b*. The view at the left of Fig. 6 is a diagram of the clutch, in which the facing is used on both sides, with springs on the periphery. It depends simply on the buckling of the plate *j* to give the necessary motion for release. It is a De Dion clutch. Some of the mechanism for the operation of the front-wheel brakes on a Belgian car is shown at the right of Fig. 6.

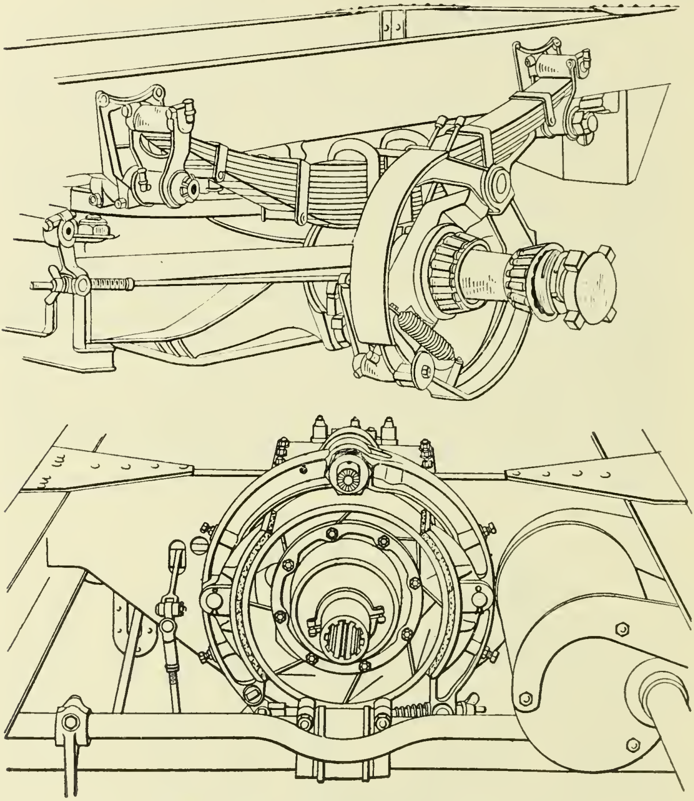

FIG. 11—TWO VIEWS OF THE BRAKE USED BY THE PACKARD COMPANY

The cam *a* is in actual line with the steering-knuckle. The cam-operating shaft is brought back and operated from the front axle. The cam *a* is spherical, so that it allows the steering of the wheel without interfering in any way with the motion of the operating shaft. The

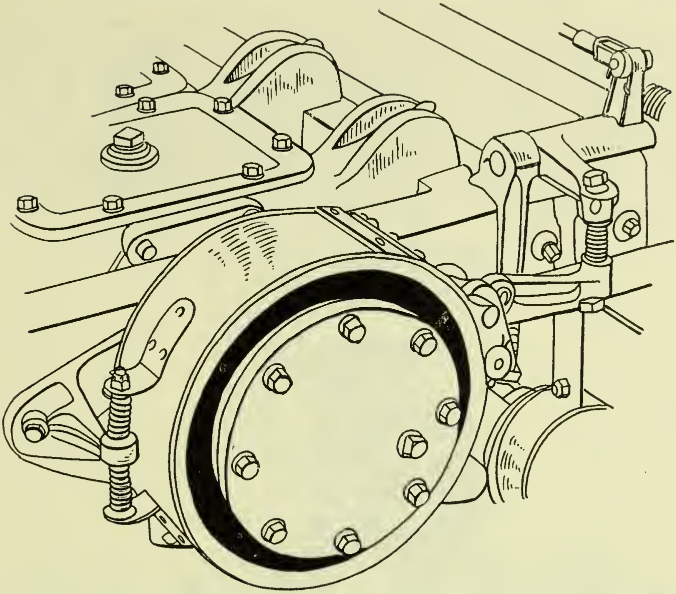

FIG. 12—BRAKE EMPLOYED ON A FOUR-WHEEL-DRIVE TRUCK

Delage and one or two of the others operate their front-wheel brakes by a cam and a flexible connection from the frame. A universal-joint connects the cam with the operating shaft, the other end of which is held on the frame by a ball-and-socket joint. This construction allows the shaft to take any angle within the limits of the springs and still give no rotating motion to the cam.

In one of the English cars the thumb or wing nuts are placed at the back of the axle so that one has access to them without very much difficulty. Fig. 7 shows an-

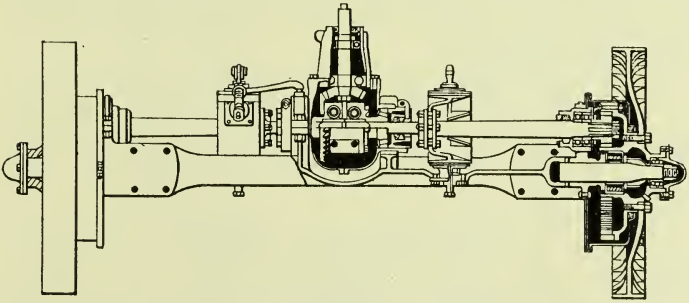

FIG. 13—AN INTERNAL-GEAR AXLE HAVING SUPERPOSED AUXILIARY OR PINION SHAFTS

other one of the brake adjustments; it is a sector of a worm wheel and worm. The worm is turned by an adjustment, drawing the rod along and so changing the angle of the camshaft. The main objection to all these devices for changing the angularity of the cam is that, by changing the leverage, a point is reached where the wear of the brake is increased and this method is no longer effective.

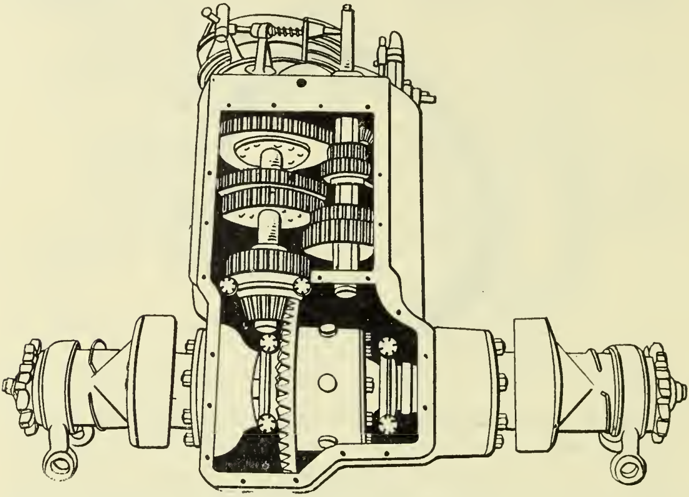

FIG. 14—TRANSMISSION FOR THE GARFORD TRUCK IN WHICH THE BRAKE BAND IS LOCATED AT THE FORWARD END OF THE COUNTERSHAFT

Fig. 8 shows a simple type of toggle brake at the left and at the right is a Timken truck brake. In the latter it is interesting to note the position of the cams; there is a certain amount of movement of the cam lever before the cam actually takes up the lost motion between the shoe and the drum.

In an internal-gear axle of the Torbensen type one of the brakes is mounted on a pinion shaft. There is a cam action that operates the brake rather than the toggle. The large steel wheels provide a good anchorage for the brake-drums.

The Russell internal-gear axle is interesting in that it represents an attempt to enclose a brake-band. It is an external brake and has an enclosure plate that comes partly round the external brake-band. This is used also to keep dust out of the internal gears, but the same com-

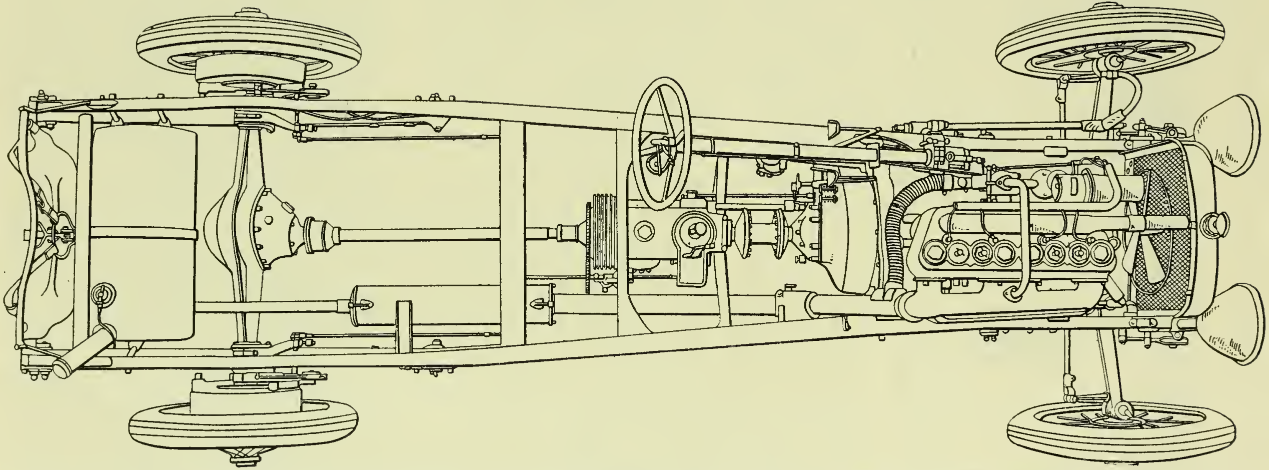

Fig. 15—A Passenger Car Chassis Having a Very Simple Brake Layout in Which the Brake Camshaft Passes through the Transmission Housing and Operates an Internal Cam on the Brake Back of the Transmission

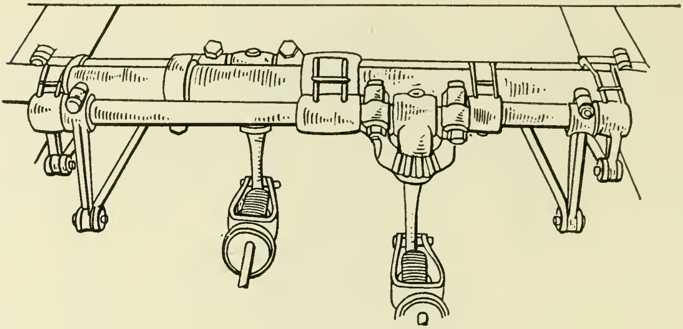

FIG. 16—A TRUCK-BRAKE CONTROL EQUIPPED WITH A DIFFERENTAL EQUALIZER

pany also uses this axle on a passenger-car type. Fig. 9 illustrates the old Knox tractor, showing the attempt made toward air-cooling of the brake-drum. Fig. 10 shows a Pierce-Arrow brake, using a radial cam, and Fig. 11 a Packard brake.

Fig. 12 shows a four-wheel-drive truck. It seems odd to have a brake-drum upon one side with nothing back of it. The chain case is immediately ahead of the brake unit. Fig. 13 is an internal-gear axle with superposed auxiliary shafts or pinion shafts; the brake units are easily accessible and are mounted adjacent to the differential case. This same construction is used on some of the Kelly-Springfield trucks. Fig. 14 is a transmission from a large Garford truck, with the brake-band up at the front end of the countershaft. There is no direct drive on this transmission, the gearing and brake being similar to those on the old Mercer car.

Fig. 15 shows the Mercer chassis, which has a very simple brake layout. It has a brake camshaft that passes through the transmission housing and operates an internal cam on the brake back of the transmission. A wheel projects out from that shaft and the connection with the clutch pedal is made to the projecting lever by a small interconnecting link. A small thumb-screw at the top forms a very satisfactory adjustment. Fig. 16 illustrates the Atterbury brake-control unit, showing the differential type of equalizer and a method of keeping lubricant on the brake shaft; it has a very accessible hand-wheel brake-rod adjustment. The Lafayette equalizer is of the differential type on both sets of brakes and is immediately back of the gearbox. The designer of this chassis prides himself on not having a crooked rod on the car.

In the Panhard chassis a steel strip, having a series of holes in its rear end, is used instead of a rod or cam. This affords a very easy adjustment at the front end. The Lancia chassis for years has had the roller-chain links around the brake-drum to secure as nearly as possible an equal contraction all the way round. This principle is employed also on the Sunbeam car which has a pulley on the end of the lever. The brakes are operated through a cable that passes around one pulley, up and over another pulley. The cable running to the other side goes round the pulleys in the same way and back. By turning a thumb-screw all slack can be taken out of the brakes; this provides an equalizing effect. The Sunbeam builder developed this method through racing. Mr. Resta told me that it was found necessary to adjust the brakes during races and, to provide a scheme whereby the mechanic could do this quickly without getting out of the car; this method was the result.

The brake lever in the Delage car comes up in the center of the car and goes forward to the front brakes. The brake cam operates at the top and the shaft is flexibly mounted on the frame and universal-joint. The Renault company has brought out a differential brake-equalizer for two brakes on the rear wheels; as well as an equalizer in which there is a worm adjustment. The brake-shaft continues through and the clutch-tube floats on it. A sector of a bevel gear that meshes with another bevel gear on the brake camshaft is mounted on one side. This brake camshaft is similar to that on the Mercer

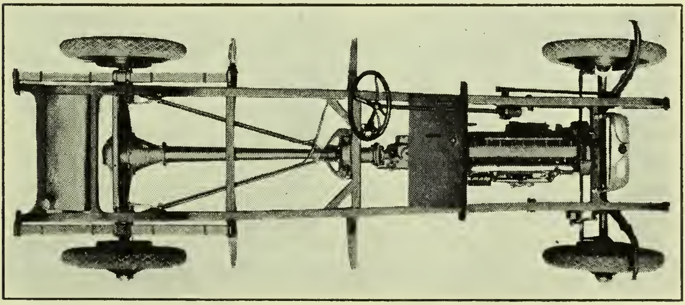

Fig. 17—Plan View of the Duesenberg Car in Which the Internal Brakes Act on All Four Wheels and Are Actuated Hydraulically

car. Fig. 17 shows a plan view of the Duesenberg car. This is another method of operating brakes, in the form of a flexible tube which has hydraulic actuation. These brakes are all of the internal type; they are four-wheel brakes.

THE DISCUSSION

W. D. Reese:—Innumerable problems must be solved in connection with the production of a safe, economical and efficient form of motorbus. Among these problems the question of brake design is certainly the most formidable. Brake failures, irrespective of vehicle type, must be vigorously guarded against, but, of course, in such cases with a bus the potential hazard is much greater on account of the larger number of persons carried.

Mr. Farwell has intimated that comparatively few improvements in brake design have been made during the last decade and that while brakes are fairly efficient, improvement in design has not kept pace with the other units in the automobile. In a general way, Mr. Farwell's statements appear to be correct. At the same time, we believe that the design of brake employed on our buses is extremely satisfactory. But this does not mean that we are unwilling to admit the possibility of improvement. As we see the situation, the fundamentals of good brake design from the standpoint of public service requirements are as follows:

(1) Safety
(2) Simplicity of adjustment
(3) Maximum service between adjustments
(4) Low upkeep-cost
(5) Freedom from loose parts and consequent rattle
(6) Ability to dissipate heat readily

During 1920 the Fifth Avenue Coach Co. carried approximately 50,000,000 passengers, equivalent roughly to half the population of the United States, and operated buses traveling 9,000,000 miles, which represents a daily mileage sufficient to encircle the earth. According to the statistics of our transportation department, this necessitated approximately 36,000,000 brake applications for passenger and traffic stops, or an average of four applications per mile. This does not include the applications made while running down grades, which would increase the total number of applications by several million.

Approximately 10,000 ft. of fabric brake-lining supplied by various manufacturers was used during the year.

We have tested a very large number of different brakes in various ways and excellent results have been obtained from those now standardized on our Model A bus. For example, tests were made with buses weighted with sand-bags to the equivalent of a full passenger load to determine the maximum braking that could be obtained with normal effort on the part of the driver. Many tests were conducted on Broadway, New York City, between 136th and 150th Streets, making runs in each direction with dry road-surface conditions, and accurate data were arrived at by a recording device consisting essentially of four electromagnetically operated pointers, a time-marker clock and a contact-making device mounted on the hub of one of the rear wheels of the bus. One of the pointers was actuated by the time-marker clock to indicate 1-sec. intervals, another by the contact-making device on the wheel to indicate the number of revolutions, and two other pointers by push buttons to indicate the length of the braking period on each brake. All stops were made without skidding the rear wheels. The results of a large number of tests showed that with

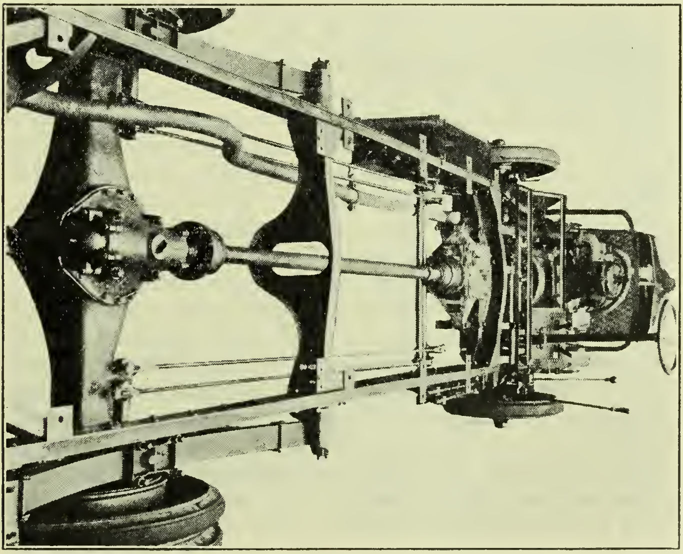

Fig. 18—Brake Used by the Fifth Avenue Coach Co. on Some of Its Buses

normal effort on the part of the driver the deceleration obtainable was 3.75 m.p.h. per sec. per sec.

Fig. 18 shows the brakes on the Model A bus. The brake-pedal is operated in a conventional fashion by the right foot and the hand-brake by the right hand. Pressure applied to these members is converted by suitable linkage through a pull on the rods leading back along the side of the frame to the cross-shaft. At this point there are hooked up four rods running to the cam-actuating levers on either side. No equalizers are used since they are unnecessary when only one point is made use of for service adjustment.

It will be realized readily that in bus work, especially with a Hotchkiss type of drive, a rather difficult problem confronts the engineer who attempts to design a brake-operating mechanism, especially when we consider the tremendous deflection and consequent axle movement that are necessary if we are to have a vehicle that rides with the minimum amount of discomfort to the passengers and at the same time assures a perfect-acting brake under either the full or unloaded condition. To take care of the deflection we use a center cross-shaft which permits of a comparatively long rod to the cam operating lever. The position of the cross-shaft and the length of the levers used have been determined as being the best combination of theory and practice obtainable after a vast amount of experimental work. To eliminate spring trouble we find that it is necessary to test all of our springs on a spring-testing machine at regular intervals. It will be appreciated that one cannot get a perfect brake action with a weak spring on one side and a comparatively stiff spring on the other.

The actual wear on the band is taken care of by setting the levers, which are placed on serrated shafts throughout the entire mechanism, and also by shims which are placed on the top of the brake anchor-plate. The brake-band is made up of a single piece of spring steel to prevent deformation. There are no joints and the ends are made perfectly symmetrical so that they can be turned upside down when the upper part is worn, this being of course the first part to wear in the wrap-up type of brake. The band is hung on a spider attached to the axle and is perfectly free to rotate, its movement being limited only by the cam. A 20-in. diameter is used with a 2½-in. width, which gives a total braking area of about 600 sq. in.

The cam used for actuating the brake is flat on top and has a radius on the bottom such that the movement of the brake-band anchor is just proportional to the pedal or lever travel. As the cam nose moves downward, it drives the band against the drum and the remainder of the braking action is accomplished by the dragging effect of the lining, which tends to intensify progressively the pressure around the surface of the band. This is proved by the fact that the greatest amount of wear comes at a point about 6 in. back from the upper brake-band anchor-plate.

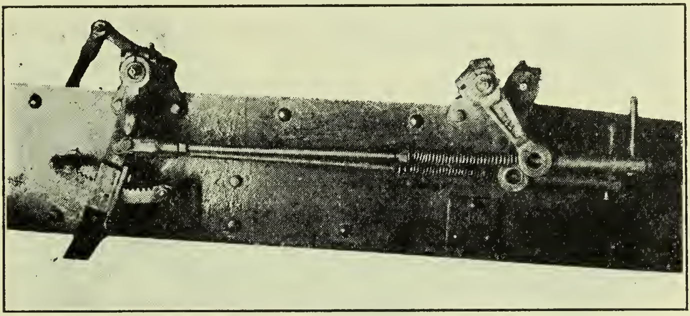

Fig. 19—A Type of Brake Adjustment That Eliminates the Use of Turnbuckles

The drum we are using at the present time is of a special-alloy cast-steel. We have made extensive tests with pressed-steel drums, but these have always proved unsatisfactory. At present we are experimenting with a heat-treated forged-steel brake-drum having a high carbon-content, which has uptodate given extremely satisfactory service.

It is interesting to note in passing that the hand-brake is arranged so that it pushes forward for application, which is just the reverse of conventional practice. The object of working the lever in this manner is that the hand has a shorter distance to travel for starting braking than it would have if one had to reach for the handle and then pull it back. This saving in time is often enough to avert a serious accident.

Fig. 19 shows a type of adjustment that eliminates the use of turnbuckles and permits road adjustments to be made rapidly and without getting under the vehicle. The tube shown, which acts as a nut, takes the ends of the rod and shortens or lengthens it as desired. It is designed so that the threads cannot possibly be damaged through carelessness. The pin, which is used as a lever, makes the tube unbalanced and consequently has no tendency to turn or change its adjustment through vibration.

The major portion of the brake adjustment is made in the garage after every 2000 miles of operation. At this time the wear on the lining is compensated for by shimming up under the brake anchor-plate so that the distance between the cam and this point of contact on the brake anchor-plate remains approximately the same throughout the life of the lining.

R. W. HASTINGS:—The firm I represent was organized to produce an improved truck-axle. To accomplish this the three factors selected for improvement were (*a*) increased accessibility, (*b*) proper enclosure and lubrication and (*c*) adequate brakes. In considering the most important features in the design of the axle, we have defined the term "adequate," as applied to our brakes, to mean a brake of large capacity, designed to deliver dependably uniform service without replacement for a period equal to the average life of the vehicle itself. Such a brake as compared with one of the band or shoe type must present an opportunity for greatly increased frictional area, an evenly distributed pressure to utilize this area fully and a complete enclosure of the mechanism as a protection from the abrasion and unreliability resulting from the introduction of foreign matter. The disc or clutch type of brake seems to fulfill these conditions best. It can be enclosed readily and, when properly designed, seems to possess qualities making it almost indestructible.

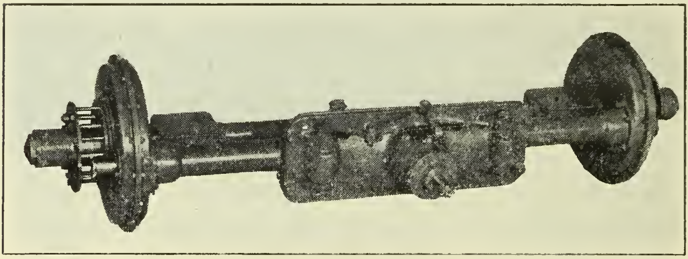

FIG. 20—A 1-TON INTERNAL-GEAR AXLE HAVING A SOLID STEEL CARRYING MEMBER WITH THE DRIVING MECHANISM IN FRONT

Fig. 20 shows our standard 1-ton internal-geared axle,

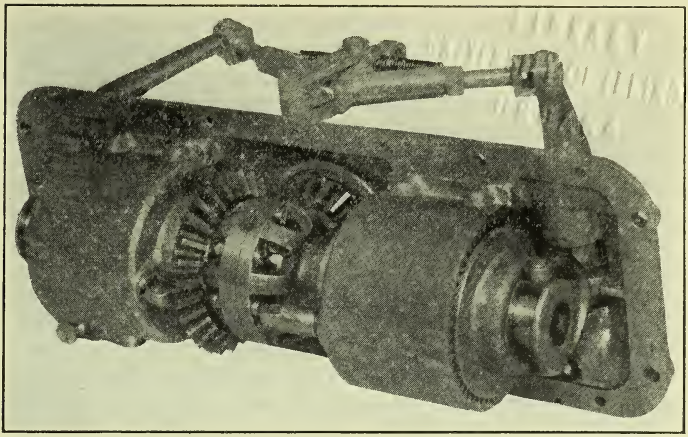

Fig. 21—The Application of the Brake to the Axle Shown in Fig. 20 Is in Two Similar Units, One on Either Side of the Differential

with the usual solid-steel carrying member and the driving mechanism in front of that member. Our internal gear is enclosed in an oil-tight case, just inside of the wheel. The pinion has jaw engagement with the drive-shaft, providing for the removal of the drive-shaft without disturbing the gears, the wheel hub and bearings or the case surrounding them.

The application of the brake, Fig. 21, to our axle has been accomplished within the enlarged differential housing, it being applied in two similar units, one of which

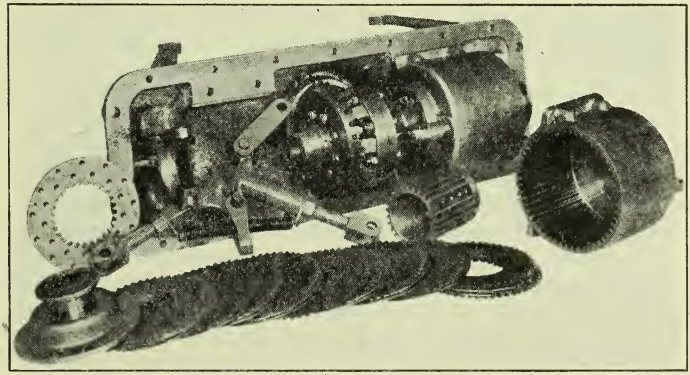

Fig. 22—The Construction of the Brake Is Very Similar to That of a Multiple-Disc Clutch

is located on either side of the differential and attached directly to the drive-shaft. The construction of the brake parallels closely that of the multiple-disc clutch, Fig. 22. A set of stationary plates of molded asbestos is slidably held within a housing that is secured to the front cover-plate. Steel rotating plates of special double construction are placed alternately with these friction plates and are slidably mounted upon the hub member, which rotates with the drive-shaft by virtue of a splined engagement with it. End-pressure is applied to the plates by a bell-shaped pressure-plate that is actuated by forked arms attached to two vertical shafts extending through the top of the cover-plate and terminating in lever arms which carry the toggle equalizing members. The toggle mechanism, Fig. 21, consists of two members having cam-shaped ends so that as their inner ends are pulled forward, thus separating the lever arms and applying the brake, the point of contact at the center between these cams does not travel forward but remains stationary, maintaining a constant angle of toggle action. This feature allows us to take advantage of the powerful toggle action and at the same time maintain a constant multiplication of leverage. While the multiplication of leverage due to this toggle construction increases the effective pressure on the brake, an item of probably greater interest and value is the automatic equalizing of the brake pressure thus accomplished. With the brake released, the toggle members are held by the return spring against a locating seat provided on the face of the cover, thus maintaining proper and equal clearance or opening for each of the brake units. Depression of the pedal draws the toggle forward, releasing it from this locating seat and thus leaving the entire system of levers free to swing to either side and so compensate for uneven adjustment. With the mechanism in this free position, it is evident that the reaction from the pressure upon one brake unit finds no resistance except that of the pressure upon the other unit, for which reason an absolutely even pressure upon each brake is guaranteed.

The question of oil circulation has been given considerable study. We have provided a means for introducing oil at the center of the brake that allow it to flow out through the specially spaced rotating plate, thus carrying away the heat that develops in the brake and giving it ample chance to radiate from the large surface of the axle housing. This provides an unusually cool brake, and we have found almost no condition under which it is not possible to place one's hand upon the axle.

Proper adjustment of the brakes is made by releasing the outer nut on the toggle cams, following this with a similar manipulation of the inner nut, setting the brake arms over and moving in the pressure plate. Enough clearance is allowed to wear out the brake without any other adjustment. We have run one job about 19 months and still have the same plates; they show almost no sign of wear.

A MEMBER:—I will relate my experience with the brakes on the Delage car from a sales standpoint. There are various advantageous factors about using brakes on all four wheels. One of the first and most important is comfort. We find that, no matter how suddenly the brakes are applied on all four wheels, there is less tendency to throw the passengers forward and out of the seats. Instead, we find a tendency of the entire chassis to sink into the wheels. In fact, one can see the hub cap sink an inch or two, as the brakes are applied harder. Another very important feature of four-wheel brakes lies in the increased safety they afford. These brakes can be applied on wet days, in snow and on ice, without chains, just the same as would be done on a dry pavement. The effect seems to be the same, with the possible exception that the brakes sometimes cause all four wheels to slide. But during an experience of 18 months they have never caused side-skidding. The general economy on brake-linings is another important item. There seems not to be the same amount of jar, because the brakes are seldom put on hard. A very slight pressure of the foot will stop the car. The economy on tires is very marked, due probably to the fact that the rear wheels do not drag. The tires seem to give much greater mileage than when using brakes on two wheels only. We have never determined the increase in mileage, but it is surprisingly large. I do not know whether that is due to the brakes or to the car, but I feel that the brakes are responsible largely. These four-wheel brakes make it very easy to drive safely in traffic. One can run up closer to the car ahead. Last and most important from a sales standpoint, we find that when a man acquires the habit of driving a car with four-wheel brakes, he is less inclined to buy one having less braking power.

On the Delage car, we find that the brakes fulfill all our requirements under all conditions. The adjustments are very simple. There seems to be no wear. With four brakes, we have double the braking surface, with half the braking effort. Altogether, we find that four-wheel brakes afford very comfortable riding and are very much favored by the public. The people who have driven cars equipped with the four-wheel brake in Europe are very enthusiastic regarding their operation.

MONTGOMERY MAZE:—The four-wheel hydraulic brake that we are now manufacturing can be considered only in the light of an accessory. Whatever we accomplish in the way of replacement equipment can be viewed only from that standpoint, for it has been necessary to work around existing conditions of design which are far from being uniform or desirable from our point of view. Future factory equipment design can readily excel in looks, efficiency and cost.

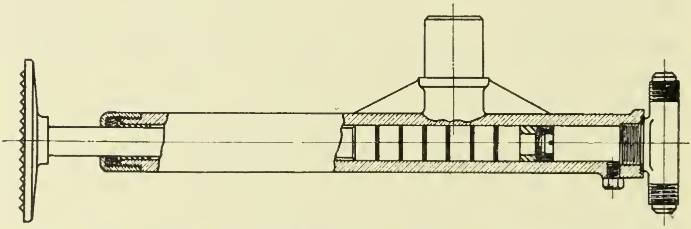

FIG. 23—PEDAL USED ON PIERCE-ARROW EQUIPMENT IN CONNECTION WITH A HYDRAULIC BRAKE

The main items increasing our efficiency are

(1) Perfect equalization, automatically obtained since the application is through fluid pressure only

(2) Complete freedom from mechanical linkages

(3) Complete absence of any effect on braking arising from relative frame and axle movement

(4) Use of 100 per cent of the car weight as a source of road friction instead of a fraction of the normal weight on the rear wheels

All four wheels are operated simultaneously by a single pedal in the conventional manner and the degree of retardation is entirely within the control of the operator. The system is purely a displacement proposition, and an emergency stop can be obtained only by the application of the emergency pressure.

Fig. 23 shows the pedal we use for Pierce-Arrow equipment; on the Cadillac the standard pedal is not disturbed. It has a normal travel of approximately 3 in.,

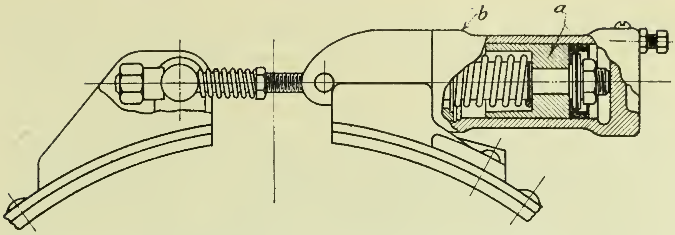

FIG. 24—CONSTRUCTION OF THE BAND MECHANISM EMPLOYED ON A HYDRAULIC BRAKE

the reserve being sufficient to wear out a third of the lining without adjustment.

Fig. 24 illustrates the construction of the band mechanism. The piston *a* is attached to one end of the band and the cylinder *b* to the other end. Pressure on the pedal displaces liquid from the master cylinder into each of the wheel cylinders alike, drawing all of the bands together with an equal force. It will be noted that no band can grip until all are in contact.

We are operating with a normal line-pressure of from 100 to 150 lb. per sq. in., but it is possible on extreme stops to set up a pressure of 750 lb. The liquid is conducted through a special, soft copper tubing, well supported at frequent intervals. This tubing is drawn to our specification and has an ultimate strength equivalent to a pressure of 13,800 lb. per sq. in.

In addition to this rigid line, three compensations are required

(1) For motion between the frame and the axle (spring action)

(2) For motion between the axle and the wheel (steering)

(3) For motion between the ends of the bands (brake action.)

The last named does not occur in our internal-brake design.

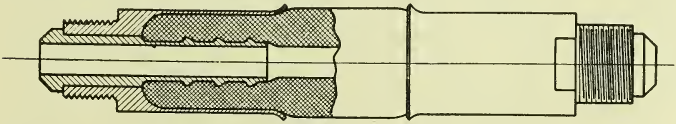

FIG. 25—SPECIAL HYDRAULIC HOSE AND A SOMEWHAT UNUSUAL FITTING ARE EMPLOYED TO OVERCOME MOTION BETWEEN THE FRAME AND THE AXLE

It has long been our intention to use rubber hose to counteract (1) but we were unable to secure a suitable fitting. This difficulty has been overcome by the development of a special fitting of exceptional merit which has made it possible to use a length of special hydraulic hose. The exposed length of hose shown in Fig. 25 is 1 in. and the flexibility is ample to compensate for any spring movement. This hose is built to our specification and each unit is tested to 3500-lb. pressure under vibrating conditions.

A swivel joint *A*, Fig. 26, is mounted directly over the knuckle-pin and in line with it to permit perfect steering. Braking cannot affect steering in any way, nor can steering affect braking. A coil pipe *B*, 40 in. long, takes up the band movement on the external brakes. This movement has a maximum of 3/8 in. The method of mounting the front-wheel equipment on an Elliott type axle is shown in Fig. 26. The swivel is mounted on the knuckles and the knuckle-pin *C*, being stationary, carries the lower half of the band. On the reversed-Elliott type of axle, the swivel is mounted on the knuckle-pin and the lower half of the band is supported from the steering arm which we replace in this case.

There are three points of closure in our system; (*a*) the copper-tube connections which are S.A.E. Standard flared-tube fittings, which are satisfactory in every way (we have never had a leak at this point); (*b*) the swivel packing boxes (these hold a maximum of special packing; because of their design that permits the operating pressures to compress the packing further, we have yet to encounter a single failure); and (*c*) the cylinder cup leathers. The last named has been a source of great difficulty but we have located a cup that has failed to show any leak during a 4-year test; with the stabilizing of the leather market, we are now able to secure a uniformity of material that obviates any further leaks at this point.

To make our system complete, however, we have adopted a standard reserve tank; it will be necessary every few months, due to slight seepage, to draw liquid from this tank. By keeping the tank full and filling the line only in this manner, all chance of drawing air into the line is done away with.

Brakes are uppermost in the mind of any owner in hilly country and he grasps at any remedy for his constant worry; but, regardless of experience, it takes but a single demonstration under any conditions to impress upon the

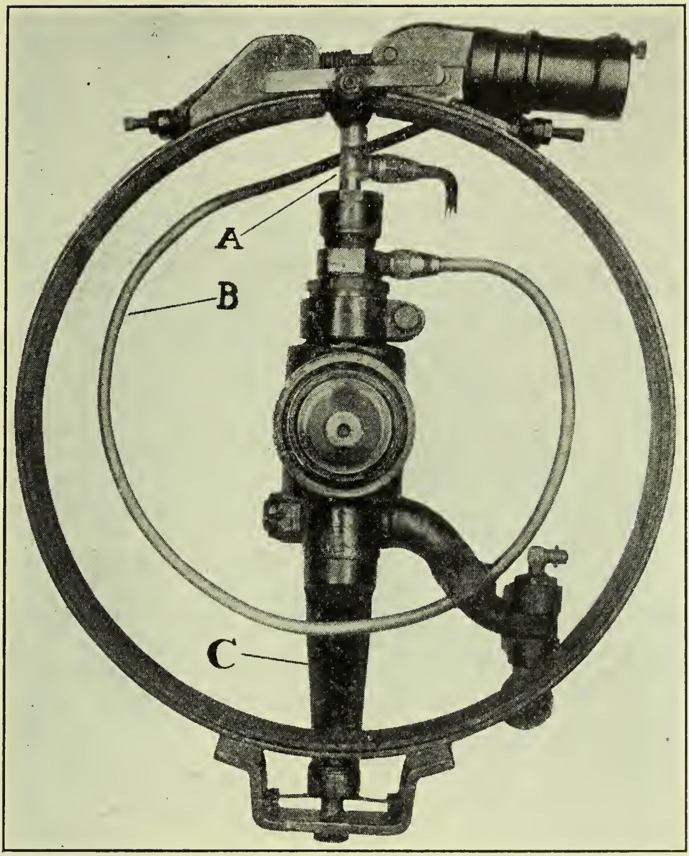

FIG. 26—DETAILS OF CONSTRUCTION SHOWING THE RELATIVE POSITIONS OF THE SWIVEL JOINT, THE KNUCKLE-PIN AND THE COIL THAT TAKES UP THE BAND MOVEMENT

operator that brakes have at last reached the plane of present-day engine design.

In addition to the safety factor given by the brakes, the owner also gets an actual monetary return. Tires can be made to show a 30-per cent increase in mileage from the standpoint of tread wear. Adjustments are not required at less than 10,000 miles, and relining of brakes is unnecessary under 30,000 miles. These figures are an average obtained from our test-cars; it is very doubtful if any owner will ever subject his car to the extreme and continued operation that they have received.

HERBERT CHASE:—Brakes must absorb power quickly when they are applied but at other times they ought not to absorb power. One common fault with both American and foreign-built cars is that brakes do drag more or less. Possibly the external-band brake drags more than the internal. There is room for better construction in brakes in general. The average brake does not compare well with the other parts of the car in the quality of its construction. British criticism of the American car with respect to the brake construction is rather caustic and, in some cases, probably is justified. It should be borne in mind, however, that British cars cost more than American cars on an average, and more expense can be put into their brake construction.

What conclusion has Mr. Farwell reached about the metallic brake-lining that is generally used abroad? There are several different kinds, I believe. I wish to know how they compare with the fabric generally used here in regard to the value of the friction coefficient. Also, will Mr. Carson describe the pressed lining that I understand he has been working with? Possibly he will describe also the testing apparatus that is being developed by the Bureau of Standards for determining the relative merits, including the wearing qualities, of different linings.

H. G. FARWELL:—With reference to the relative coefficient of friction of the metal brake-lining and the fabric lining, there seems to be considerable discussion and difference of opinion. The coefficient of friction will run approximately 0.4; with a bronze shoe it will run approximately 0.2. That has been corroborated by engineers from abroad. They use more encased drums abroad than we do here; pressed-steel drums are used almost altogether.

V. W. PAGÉ:—I witnessed some tests of the multiple-disc brake. I was afraid there would be considerable drag and attendant heating. After a number of tests down a steep test-hill, I was able to put my hands on that casing without any discomfort; a standard touring car that accompanied the test car stuck on this same test. I could not place my hand anywhere near the brake-drum on account of the heat. Then we tried some coasting tests. We find fully as good results with that form of multiple-disc brake as would be obtained with the conventional band-brake well adjusted, and considerably better results that one would get with a band-brake ordinarily adjusted.

C. Carson:—Mr. Chase has requested information regarding the testing apparatus at the Bureau of Standards. The Parts and Fittings Division of the Society's Standards Committee was assigned the subject of brake-lining and it undertook to develop some standard method of testing brake-linings in collaboration with the Bureau of Standards at Washington. The results so far obtained have not entirely solved the problem, but very gratifying progress has been made.

A pressed-steel drum was mounted on the shaft of a dynamometer and a skeleton frame of the general form of a prony brake was built around it. A spring-balance was installed between the arms, to adjust the load on the brake-shoes. Two short flexible bands were used instead of a complete encircling band, as is commonly found on wheel brakes. These carried linings about 11 in. long, 2 in. wide and ¼ in. thick, and the pressure was applied so that a nearly uniform pressure per square inch would result. First, we tried to find what pressure per square inch could be carried and what velocity should be used. To accelerate the test, an attempt was made to run the apparatus with a water-cooled drum. That was provided by putting a plate equipped with the usual tube for introducing water in the open end of the pressed-steel drum. But when using high pressures and high velocities, heat is generated so much faster at the point of contact with the lining on the drum than it can be transmitted through a steel drum ¼ in. thick that, even with a drum containing water, the surface of the drum will fuse while it is running. So, we were forced to abandon the theory of running at high pressures with a water-cooled drum, because the water dissipated only a limited amount of the heat generated. The tests are not yet complete. We have made a long series of tests using both water-cooled and dry drums. Apparently, a character of lining that will give excellent results at moderate loads will break down and give very unsatisfactory results when high pressures and continued brake applications are given to it.

It seems unfortunate that as yet, all through the industry, there has been no standard method of testing such materials. Some firms have tested brake-lining by making what might be called a skid test. They apply a piece of friction material to a rotating drum, hang a certain weight on it, make it turn for a certain number of hours at a certain number of revolutions per minute and record the result. The material that endured the longest was given the credit of being the best, without taking into consideration the power absorbed by the brake during the run. We have tried to eliminate such a condition in the apparatus we use. At present, the doing of a uniform amount of work is taken as a basis for the test. The pressures per square inch are varied to make the power consumption constant at all times. We keep the revolutions of the dynamometer and the power consumption constant and change the pressure on the lining. We feel that, if the linings are doing the same amount of work, we can then approximate a fair comparison of their life.

Among other interesting results, we found that some of the yarn in the linings had been made with brass-wire cores and that the surface of the lining became covered with copper plating; to a certain extent the steel drum was coated likewise. Investigation indicated that the heat generated was sufficient to drive the zinc out in the form of vapor or dust. We have found that the degree of vulcanizing in the rubberized linings seems to have a very pronounced effect on their wearing quality. Linings made from the same fabric and having practically the same rubber mixture, but different degrees of vulcanizing, will vary in their performance under the same load conditions from 20 min. for one sample to 16 hr. for another.

A recommendation probably will be made to the users of brake-linings, asking them to modify, if possible, their method of installation. A prevalent method of installing lining requires the strip to be very flexible as the usual practice is to rivet it at the ends first and then press out the kink left in the center to make the lining hug the band tightly. That requires a considerable flexibility in the material. We believe that in adhering to that flexibility the users of lining are sacrificing much of the life of the lining. That is indicated by the change in wear according to the degree of vulcanization. With a vulcanized or with the woven type of lining, hard pressed and impregnated with a hard compound, this kinking method of installation could not be used and there would have to be a change in the method of application.

We find that they are using very hard cured lining in European practice. The Ferodo lining, which has a corrugated shim between the lining material and the band to circulate air and dissipate the heat, is an example of such material. This lining is an asbestos woven fabric; it is very hard, compressed to a very high degree and almost lacking in flexibility. It is installed usually in comparatively short curved pieces, because it is not flexible enough to bend around a band. My opinion is that, for long life and maximum service, the present method of installing brake-lining and its degree of hardness must be changed.

COEFFICIENT OF FRICTION OF BRAKES

The determination of the coefficient of friction is perhaps the most elusive problem we deal with. We have not even been able to determine it as a constant on any one particular sample during a run. We doubt very much if a really fixed coefficient of friction can be maintained with an impregnated, woven, or folded and stitched fabric, or for any fabric composed of yarns interlaced. It may be reached in some new form, such as unwoven or pulp lining commonly called molded material, used in some types of clutch-facing. A rough value for this coefficient would be about 0.40, but I think possibly that it should be modified to 0.36 for a woven and 0.42 or 0.43 for a rubberized lining, the latter having a slightly higher coefficient. Almost any coefficient desired can be obtained by changing the compound. One can make a lining having a severe grip or, changing the compound by introducing certain other ingredients such as waxes in one form or another, secure a low coefficient.

From the tests, we believe that the reason for the variation in the friction coefficient is that at no two times during the wear of a piece of lining is there the same condition of surface contact. Consider a piece of folded and stitched, laminated lining with a rubber compound. At the beginning, there is a veneer or surface of rubber that has a certain coefficient of friction on the steel drum. As the wear progresses, the coefficient changes because the surface contact is composed of a certain area of asbestos fiber, metal and rubber. The areas of the three materials in contact change continually and the coefficient of friction changes correspondingly. While there is no uniform cycle of performance, there is a fluctuation and continual variation, even in the same piece of lining.

L. G. NILSON:—Who has had any experience with the metallic brake-lining that is a composition of lead and copper?

MR. CARSON:—We have not tested any of that material. I saw some clutch-rings that were made of that material recently. The engineer who conducted the test of the rings eliminated that material on account of its high cost and because the coefficient of friction was so low that it would have been necessary to increase the area of the clutch to a prohibitive amount to use it interchangeably with asbestos materials.

W. C. MARSHALL:—What tests have been made to show which type of brake is freer from oil, the internal or the external? The efficiency of the brake depends largely on whether the oil gets in on the brake-band and the drum. In some cases the oil might be thrown off. In other cases it might hold.

MR. FARWELL:—Our experience shows that oil gets in on both types. Probably a greater amount of the oil will be retained on an internal than on an external type. We find, in some cases, oil or grease on both the internal and external bands. It seems to involve choosing the lesser of two evils.

N. G. BERGENHOLTZ:—In regard to the brakes on the buses of the Fifth Avenue Coach Co., it seems that many of these are not shoe brakes and do not act equally in either direction. On this particular brake, our attention was called to the fact that the cam was not of the same shape on both sides. How does that brake act in going in a reverse direction? Does it give an equal braking effect?

MR. REESE:—No, it does not. The brake that we use is known as the "wrap-up" type. The efficiency is greatest when going in the forward direction. When going in the reverse direction, it is much harder to apply. I should judge that if one could apply the necessary pressure the efficiency would be the same.

MR. BERGENHOLTZ:—Is the object of that cam shape only to take up the slack first, before the pressure is applied, rather than to try to equalize the pressure in both directions?

MR. REESE:—Yes.

M. C. HORINE:—A self-wrap brake is essentially a one-direction brake. Brakes have been developed which are known as the double-wrap type but a double-wrap brake, either external or internal, is practically not a wrap-type brake at all. That is, it is not a snubber; it does not work on the principle utilized when several turns of rope are taken round a capstan, as is the case with the ordinary self-wrap brake, because the self-wrapping on one side is compensated for by the unwrapping of the other. This matter of having a brake act equally in either direction is important, particularly in motor-truck work; great weights must be considered and gear-changes are not so certain on a grade, because the truck is going at a slower speed and its inertia will carry it forward a very much shorter distance. The experience of the company I represent was such that, previous to the time that type of brake was abandoned, it was necessary to redesign the cam so that it had equal action on both ends of the band. The only advantage of the flexible band in a double-wrap brake is to give a more or less equalized pressure.

The effects of pressure and speed on the wear of a brake have been suggested. The amount of wear on a brake should be roughly commensurate with the amount of energy dissipated. It seems that a brake might be designed with a small drum operating under very high speeds. This would mean lower friction at higher speeds and no more wear per square inch than with a larger brake operating at lower speeds at higher pressures. It seems to me that the wear would be dependent upon the amount of area. There is considerable buncombe with regard to braking area. It is possible to design a brake with a great braking area, much of which is worse than parasitic. Taking the shoe illustrated in Fig. 27 as an example, it would be possible to line it up to the tip where the cam contact is and down to the hinge. Such a brake acts as a lever. The portion of the shoe at *a* is on the wrong end of the lever. One can get very little pressure at that point, whereas at *b*, close to the fulcrum, one can get a great amount of pressure. If the portion *a* is lined, the lining will act as a spacer. It is impossible to get sufficient pressure on it to make an effective brake, and yet it acts as a spacer to prevent the portion *c d* of the lining, which produces effective braking, from making contact. The portion of the lining at the point *b* does not reach the drum after a certain amount of wear, because it is so close to the hinge; so, it is largely parasitic area. On a rigid drum of this character I believe it is not necessary to have a lining for more than about the distance *c d*. I do not believe that any more lining on that shoe will give braking effect. In a case where it is close to the tip, it may prevent effective braking.

In regard to having internal and external brakes on the same drum, which is the conventional practice on touring cars and the cheaper kinds of truck axle, it does not seem right to me to put two brakes of any type on the same drum. Asbestos is hardly a good conductor of heat, whereas iron is a very good conductor. Since the drum is the brake member that contains most of the heat, it seems reasonable that the binder in the lining should burn because the drum gets hot. If one could always apply a brake to a cool drum the lining would not burn. Suppose we have two sets of brakes which we apply alternately in descending grades to avoid burning either set. If we apply the alternate sets of brakes to the same drum, which has already become heated by the application of the preceding set, the second set will burn almost immediately. Hence, the ideal brake arrangement would be for each set of brake shoes to act on a separate drum, which is the condition we have with four-wheel brakes and with shaft brakes.

Another consideration in connection with shaft brakes is that of equalization. It is possible to equalize the pressure on two brakes in a number of ways, such as using a simple cross-tree, a differential arrangement, pulleys, fluids or other means. But that only equalizes the pressure and does not equalize the braking effect. It appears to me that the only way to equalize the braking effect is through the differential, inasmuch as two tires will never follow exactly similar tracks and it is the tire on the road that actually gives the braking effect. That seems to me a strong argument in favor of the shaft brake. However, the shaft brake has some defects which I think have not been given sufficient attention.

The greatest complaint against shaft brakes is that they chatter. Chatter in the shaft brake is due to many causes, chief among which is the fact that the brake

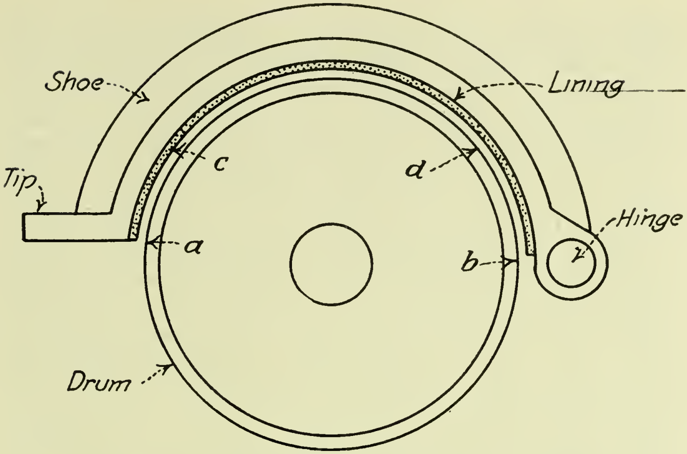

FIG. 27—AN EXAMPLE OF FAULTY BRAKE DESIGN

itself is not supported firmly enough. Most shaft brakes overhang on a propeller-shaft and, naturally, since there is no propeller-shaft that remains concentric, there is a slight wabbling of the drum which ordinarily is supported separately from the shoes. Another cause of chatter is the looseness with which the actuating means and the shoes themselves are attached. If the shaft brake is properly designed with rigid shoes and a drum that is mounted between bearings, as one would hang a grindstone, and if the entire brake and its mechanism is supported by one rigid frame, there will be no chatter. I know this because I have experimented with brakes of that character.

The ordinary method of mounting a shaft brake is to have it operated by a pedal. A number of cars now obsolete had brakes of that sort, and it was characteristic of the operators of those cars that they almost never used the foot-brake. The operator always used the hand-brake because the shaft chattered. If it must chatter, the shaft brake should be operated by the hand-lever, because ordinarily the hand-lever is the one which is used to lock the car when parked. This means that it is generally applied when the car is stationary. The foot-brake ordinarily is used only when the car is moving; therefore, if we must have a chattering brake, let it be the hand and not the foot-brake. Another reason why the shaft brake should be operated by hand is that spring action does have the effect of shortening and lengthening brake-rods that are connected to the rear axle, and that it is a very common experience when the brake is applied with the car loaded to have it release itself when the car load is taken off. With very light cars, where the hand-brake acts on the rear axle, that is a common experience; when the brake is applied while the passengers are in the car, it releases itself when they get out of the car. That is experienced to a much greater extent on trucks. It is not exactly a common experience, but it does happen occasionally that a truck releases its brakes when the load is taken off, the brakes having been properly locked while the load was on. Another common result is that when the hand-brake has been applied with the truck empty it becomes impossible to release the brake after the truck has been loaded. Naturally, a shaft brake, fixed to the frame cannot be affected by spring deflection and hence is the ideal hand-brake.

A certain amount of prejudice against the shaft brake originates from the fear that, acting through the drive-shaft, universal-joints, drive gears, differential and axle shafts, it is less reliable and that the parts will be subjected to an abnormal stress from a sudden brake application. Experience shows, however, that failure of rear-axle brakes due to crystallized brake-rods, stuck and rusted pins and burned-out linings is more common than failure of driving parts. The strains to which the driving parts are subjected from shaft brakes, furthermore, are not so severe as is commonly supposed. It can be demonstrated easily that the shock on these parts produced by a sudden application of the clutch at high engine-speed with the gearshift in low or reverse of our modern large-range gearboxes greatly exceds the brake torque at which the wheels will slide.

A solid rod is apt to vibrate and crystallize. Cable is one of the first means we used for applying brakes to automobiles, but this has never been entirely satisfactory because cable is apt to fray. The flat-steel strap seems to have a certain amount of promise, except that it is a single strand and, as was found in airplane practice with streamlined solid wire, it is hardly safe. Strap of that sort is brittle and, if it fails, it fails clear across and the rod is broken. Some time ago I had a different style of brake-rod or cable on a small car, that consisted of a form of chain made up of flat brass links, each link being folded so that the holes in the free ends registered and permitted the next link to be folded through these openings. The experience I had with that substitute was very encouraging. I think a little experimenting in the use of these folded sheet-metal chains will show that they have real possibilities as substitutes for brake-rods. They are extremely flexible, can be made very strong, and offer a very ready means of adjustment.

With regard to the disc brake housed in the differential housing mentioned by Mr. Hastings, granting that it does have a very powerful effect and very long wear and that the brake keeps cool, is not that effect at the expense of the oil? Is it not true that the oil in that housing deteriorates very rapidly because it is being burned between the brake surfaces?

MR. HASTINGS:—We have operated this brake about 19 months and it was our practice to remove the oil frequently for inspection. We found that it did not deteriorate. During the last part of the run we have had one supply in the housing for 4 or 5 months and we find it in good condition today.

MR. HORINE:—Can you account for that?

MR. HASTINGS:—It is because there is a very large area and the heat is distributed over that area.

MR. HORINE:—Will not particles of brake facing, dislodged by friction, be circulated with the oil and do considerable mischief in the gears and bearings?

A. M. WOLF:—It is one of the first fundamentals of brake design to have them absolutely free when they are in the "off" position, and a correct brake should absorb absolutely no power when not in use. Dragging brakes are a prevalent failing and obviously a large factor in fuel consumption.

To mention a few other means of braking, we can use air when coasting downhill, with the switch off and working the engine as a compressor. However, the ordinary engine is not a very efficient brake under these conditions. The Saurer engine is built with a sliding camshaft that modifies the valve action so that the engine will absorb a maximum amount of power while turning over. The Saurer people also tried out a fan brake. The fan was mounted under the center portion of the chassis where it had plenty of room and, when it was thrown into action, its resistance was similar to that of a fan dynamometer absorbing energy.

An hydraulically actuated brake was mentioned, and attention has been paid also to hydraulic means of absorbing power. A car was developed in Europe in which the constant-mesh gears of the transmission were encased so that they would form an oil gear-pump. To cause braking effect, a pipe through which the oil circulated was blocked by closing a valve interposed therein. I understand, however, that this was not a success, due to the excessive heating of the oil and the very high pressures encountered; but with modern methods this idea might be revived. Most hydraulic transmissions function as a brake when the control valve is shut or set in a position corresponding to a speed slower than the prevailing one.

The airbrake that uses air as an actuating medium is somewhat old, having been applied on the first Northern four-cylinder car. This car had an air clutch, as well as brakes applied by air-actuated pistons operating the brake-rods. The clutch consisted of a large leather disc forming part of a bellows and was mounted so that, when air was admitted behind it, it would extend slightly forward and come into contact with the rear finished face of the flywheel. The airbrake is now being applied to trailers; it seems that the trailer application will cause both hydraulic and air actuation to become popular. It will be recalled that the Knox-Martin tractor was brought out with an hydraulically actuated brake.

Small reversible high-speed electric motors, acting through a large worm-gear reduction, have been used to actuate the brake-rods. A small button or lever switch-control makes the operation extremely simple; but such a system involves many complications in performing an operation that can be accomplished by very simple means. Unusually large vehicles might be an exception.

With reference to brake adjustments, I desire to mention a device which automatically tightens the brake-rods when their travel is too great. This is done by a ratchet that is held in fixed relation from a cross-member. Movement of the brake-rod beyond a predetermined limit causes the ratchet to rotate a member which is threaded over the brake-rod. This device is borrowed from railroad practice; the slack adjuster, in this case air-actuated, is located on each brake cylinder.

The mounting of the brake cross-shafts deserves consideration, so that they shall be free from binding due to distortion. It was interesting to see how the Renault design obviated any such tendency by its universal mounting. There is one truck on the market that has a cast cross-member which also forms an anchorage for the front end of the rear springs. All the brake cross-shafts are mounted on this member and, due to its unit assembly, there is very little or practically no chance of binding occurring in service.

It is interesting to note the disappearance of the one-time long equalizer bars. They often exceeded the frame width, so that the rods running to the drums would be outside of the frames. Today, the rods are being kept within the frame side-rails. Truck design also is reverting to this method to a large extent. It allows a more substantial brake rigging on the axle.

In the Hotchkiss drive, due to the displacement of the axle under torque, driving and braking stresses, some builders allow for a certain amount of lost motion between the pedal and the final brake-rod. Considering the internal brake, I believe that we should design cams with a small circumferential section or base circle before the shoes are expanded. It naturally would be a more costly cam. This is not necessary, of course, when radius-rods or an anchored torque-tube is used, if the clevis-pin of the brake-lever has the proper location.

Regarding the lubrication of the brake rigging, we see cars today with grease or oil-cups in places that the owner or driver will never bother to reach. In fact, some cannot be reached without crawling under the car on one's back. This refers to brake cross-shafts on the frame, and also to brake shafts on the axle. I am a firm believer in the oilless bushing, of any of the several types, for such locations and I am surprised that all companies do not use them.

CHASSIS FRICTION LOSSES[1]

By E H Lockwood[2]

The loss of power due to the friction of the various parts of the chassis has been carefully and elaborately investigated by a dynamometer, the dual purpose being the determination of the amount of internal frictional resistance of the front or rear wheels and the measurement of the power that can be delivered at the rear wheels with the concomitant rate of fuel consumption.

The rolling-friction due to the resistance of the wheels as a whole is taken up first and afterward the separate resistances of the tires, bearings and transmission are studied under varying conditions of inflation-pressure and load. The five frictional resistances that were chosen as giving the most useful information are those of the front tires, the rear tires, the front bearings, the rear bearings and the engine.

Among other topics considered are the ratio of the total friction loss to the weight of the vehicle; a comparison of the resistance of passenger cars and trucks, of solid tires with pneumatic and of fabric tires with cord; the ratio of tire-friction to bearing friction; the rules for determining the total friction of the chassis; the effect of variations in the load and the inflation-pressure on the rolling resistance of pneumatic tires; the development of resistance formulas for fabric and cord tires; a comparison of the wear of pneumatic tires with that of solid tires under the same load; the effects due to variations of the speed and to heavy-wall tubes, roughness of the tread, non-skid surfaces, large and small sizes, age and duration of wear; the development of heat in the tires; and the influence on friction of the rise of temperature of the gearbox lubricant.

The apparent decrease of friction during the last few years and the uniformity of the products of certain manufacturers are noted. The results of the tests are shown in detail by numerous charts and tables.

The dynamometer drums consisted of metal-shrouded paper cylinders, mounted on a heavy shaft hung from the basement ceiling on ring-oiled babbitt-bearing hangers. The tops of the drums projected slightly above the main floor through openings in the concrete.

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[1] Metropolitan-New England Sections paper.

[2] M.S.A.E.—Assistant professor of mechanical engineering, Yale University, New Haven, Conn.

The diameter was about 67 in. the faces were 15 in. and the overall width was 71 in.

Dynamometer measurements of a chassis were made by placing one pair of wheels on the tops of the drums, so that the wheels and drums revolved in rolling contact. Two distinct ends were sought: (*a*) the determination of the amount of internal frictional resistance of the front or rear wheels and (*b*) the measurement of the power that the engine can deliver at the rear wheels, and the rate of fuel-consumption.

The power to drive the drums was obtained from a 15-hp. variable-speed electric motor which was belted to the drum-shaft. The power delivered to the drum-shaft by the engine of the car was measured on a prony-brake pulley of 100-hp. capacity. The drum-shaft rotation was measured by two ratchet counters and by an electric tachometer reading in revolutions per minute. All the apparatus could be observed and controlled from an operator's table on the main floor near the drums.

Fig. 1 is a view of the dynamometer from the basement, showing the driving motor and the prony brake. Fig. 2 shows the operator's table with the starting rheostat and meters for the electric motor, drum tacho-

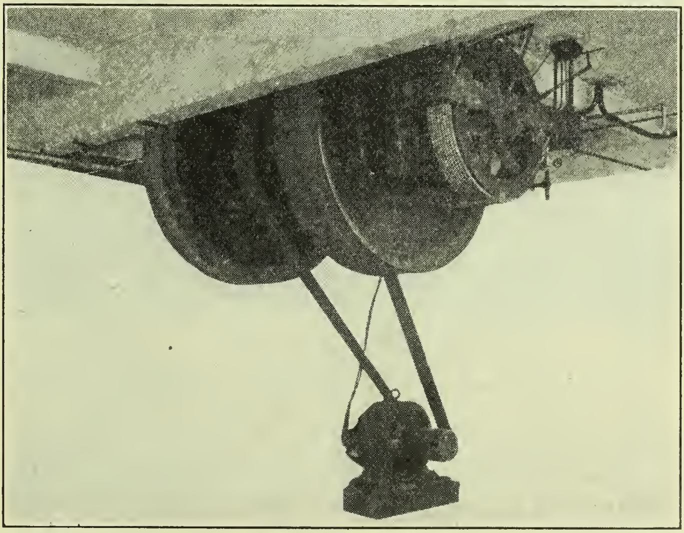

Fig. 1—View of the Dynamometer from the Basement Showing the Prony Brake

meter, revolution counters and scales for weighing the prony-brake load. The entire apparatus is shown in vertical section in Fig. 3 to which reference may be made by the following letters:

a, a,	Main drums
b	15-hp. variable-speed electric-motor belted to the drums
c	Prony-brake pulley on the main shaft
d	Electric tachometer, geared to the main shaft
e	Indicating dial of the electric tachometer
f, g	Direct-current meters for the 15-hp. motor
h	Starting rheostat for the 15-hp. motor
i	Platform scales for weighing the pull on the brake-arm
k	Control-handle for adjusting the brake-band tension

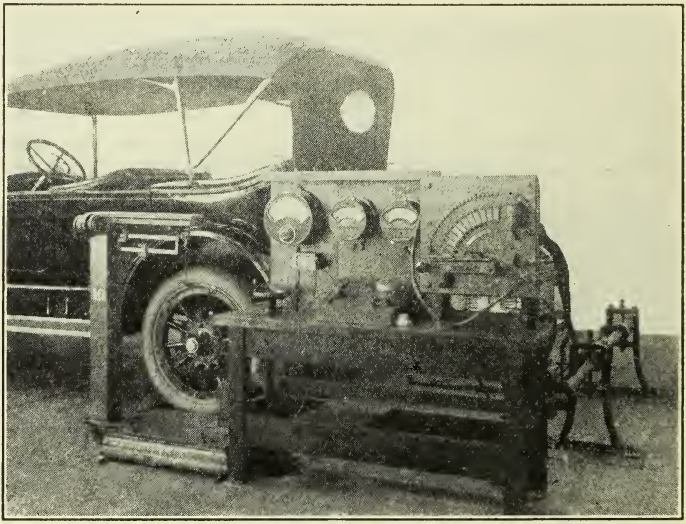

FIG. 2—OPERATOR'S TABLE

MEASUREMENT OF ROLLING FRICTION

The simplest measure of rolling friction is the tractive force that must be applied to the axle of the wheel to overcome the resistance. The tractive force can be measured by a spring-balance when the wheel is being towed, as shown in Fig. 4 at the left. The towing method is not practicable except at very slow speeds. The drum method, shown in the right-hand portion of Fig. 4, pro-

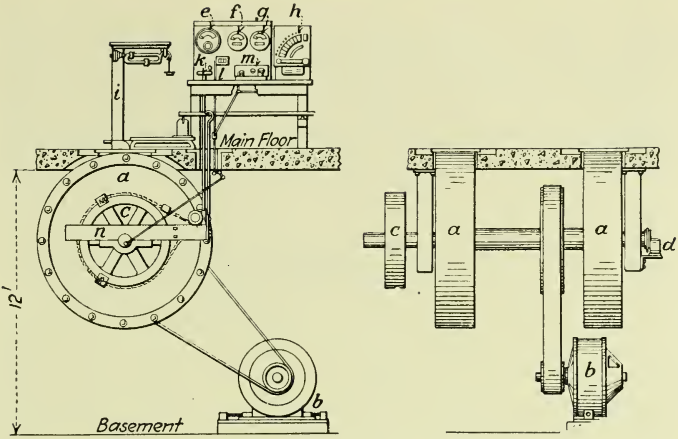

Fig. 3—Vertical Section of the Apparatus

duces the same result as the towing method, but avoids its drawbacks. In this case the wheel rotates with the axis stationary, while the tractive force required to revolve the drum is being measured. The tractive force at the drum circumference can be measured conveniently and accurately by the dynamometers on the Mason Laboratory drums. The rolling friction, or rolling resistance, is measured in pounds, and is the tractive force required to revolve the drums against the resistance of the wheels. The measurement of rolling resistance requires two separate steps: (*a*) the drums are revolved with the wheels in place, while the electric input is carefully read and recorded; during this stage, the current

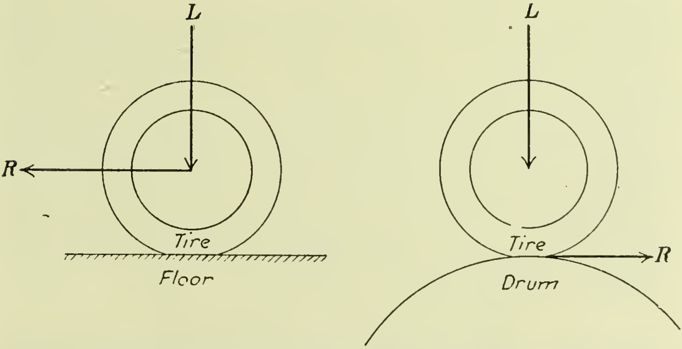

Fig. 4—Diagram Showing the Towing and the Drum Methods of Measuring Rolling Friction

input represents the total power required to drive the drums and the vehicle wheels; (*b*) the wheels are removed from the drums, after which the idle drums are rotated at the same speed as before. The operator then adds the load to the prony brake until he has duplicated the original readings of the electric meters. When this has been done, it is plain that the *added load* in the second step is equal to the *subtracted load* in the first step, due to the removal of the wheels. In other words, the added load on the prony brake, expressed in pounds at the drum circumference, is a measure of the rolling resistance.

This method can be varied to give the tire resistance alone, without the resistance of the bearings. To do this the wheels are not removed from the drums, but are jacked up until the rubber surface rests lightly upon the drums with contact enough to cause the wheels to revolve. The second step is then repeated, load being added to the prony brake until the original readings of the current meters are duplicated. The increase in the prony-brake load in this case will be found to be smaller than before, since it represents not the total resistance of the wheels, but only that portion of it that is due to the flattening of the tires on the drums. In jacking up the wheels for tire-resistance measurements, no definite pressure between the tire and the drum is required. Tests prove that any light pressure will suffice, provided it does not cause appreciable flattening of the rubber.

This method was adopted because of its simplicity. It has proved sensitive in practice and readily allows small variations to be detected when the inflation-pressure or the load is changed. The method, however, involves a slight error for which no allowance has been made. This lies in the assumption that the drum-bearing friction and the wheel-bearing friction remain unchanged when the wheel is jacked up for the second reading. It is evident that the bearing friction must be less when the wheels are jacked up, hence the prony-brake reading represents not only the tire resistance but also the slight change in the bearing resistance. The final result, therefore, is slightly too large. It has been deemed safe to neglect this error, since it is very small and does not affect the validity of comparative conclusions.

TOTAL FRICTIONAL RESISTANCE

Friction measurements have been made at intervals since 1916 on cars having great variations of weight and tire equipment. About 50 typical examples have been chosen for the comparison of the ratio of the total friction loss to the weight of the vehicle. These cars had pneumatic tires and ranged in weight from 1800 to 5300 lb. Fabric tires were used on the lighter cars, in most cases, and cord tires on the heavier ones. The inflation-pressure was from 60 to 80 lb. per sq. in. Observations were made at speeds of 20, 30 and 40 m.p.h. The results were averaged because the friction was practically constant at all speeds.

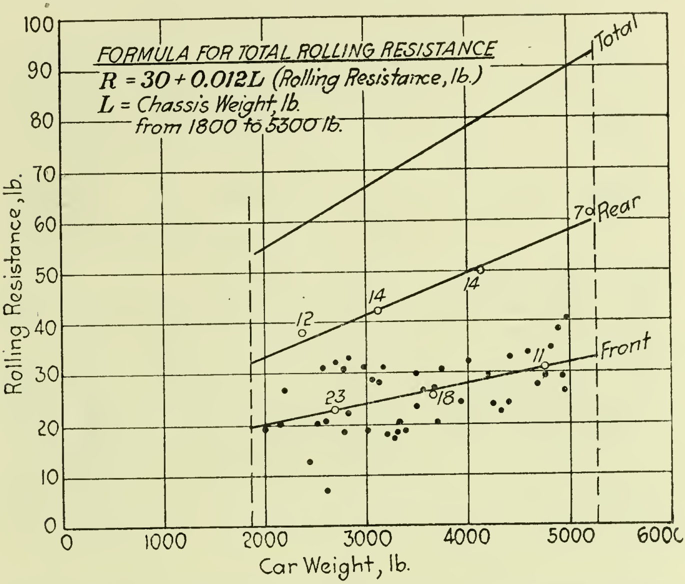

Fig. 5—Chart Giving the Average Results of Tests To Determine the Rolling Resistance of the Chassis of 50 Cars Equipped with Pneumatic Tires

The front-wheel and the rear-wheel resistances were measured separately, and the results were plotted with the weight as the base line. Variations were found in cars of the same weight, as might have been expected. These are shown in Fig. 5 by the dots surrounding the line marked "front." An average line was plotted by taking the mean of the results for a group of cars having similar weights. These points are indicated by the encircled dots, the number showing the size of the group. A straight line was found to represent the tractive frictional resistance of both the front and the rear wheels of the car, after which they were combined into a single line marked "total."

The total frictional resistance of the average car as shown by these tests can be represented by the simple formula

$$R = 30 + 0.012\,L \qquad (1)$$

where

L = Total weight of the vehicle, in pounds
R = Total frictional tractive force, in pounds

The results given by the formula and the diagram in Fig. 5 refer to the total frictional resistance of the tires, bearings and transmission on a smooth road. The resistance due to the wind is not included. The results given by the formula are for the average car and a considerable variation from this average can be expected. An easy-rolling car with cord tires may have, perhaps, 15 lb. less resistance than that given by the formula, while a hard-running car may have more.

The frictional tractive force of vehicles frequently has been expressed in the special unit "pounds per ton." This unit is convenient when the tractive force varies in direct proportion to the weight of the vehicle; otherwise it is not convenient. For example, in the group of 50 cars referred to, the "pounds per ton" varied from 54 to 34. This unit is somewhat awkward and open to ambiguity, since the ton has two recognized values, 2000 lb. and 2240 lb., one being used mostly in America and the other in England.

It seems proper to suggest a new unit that is not ambiguous, namely, "pounds per thousand pounds," or "pounds per M." This unit is merely one-half the pounds per ton when the net ton is used. For example, 40 lb. per ton is the same as 20 lb. per M. Moreover, the new unit is directly comparable with the usual coefficient of friction by changing the decimal point. Thus 20 lb. per 1000 lb. becomes 20 lb. per M; and the frictional coefficient is therefore 0.020.

RESISTANCE OF TRUCKS

The frictional resistance of a number of heavy trucks equipped with both solid-rubber and pneumatic tires has been measured. These results, shown in Table 1, are interesting for comparison with passenger cars. The trucks have additional interest, as they belong to a group that is now being used for the measurement of tractive resistance under actual road conditions. The road tests

TABLE 1—FRICTIONAL RESISTANCE OF TRUCKS

Specification	Quartermaster Standard Type B Truck		Mack Chain-Drive 7½-Ton Truck
	No. 432,799	No. 44,913	
Front Wheels			
Weight, lb.	4,665	4,395	5,200
Tires, Solid Rubber Single, in	36x5	......	36x7
Tires, Pneumatic Single, in.	......	38x7	......
Total Tractive Force, lb.	80.0	63.0	98.0
Tractive Force per 1,000 Lb. of Weight, lb.	17.5	14.4	19.0
Rear Wheels			
Weight, lb.	6,875	6,300	7,115
Tires, Solid Rubber Dual, in.	40x6	......	40x7
Tires, Pneumatic Single, in.	......	44x10	......
Total Tractive Force, lb.	155.0	215.0	195.0
Tractive Force per 1,000 Lb. of Weight, lb.	22.5	34.0	27.5

Loaded Truck, Complete

	Weight, Lb.	Total Tractive Force, Lb.	Tractive Force per 1,000 Lb. of Weight, Lb.
Type B Truck, No. 432,799	11,540	235	20.3
	14,540	292	20.0
	17,540	344	19.6
	20,540	396	19.3
Type B Truck, No. 44,913	10,695	278	26.0
	13,695	349	25.5
Mack 7½-Ton Truck	12,315	293	23.8
	15,315	340	22.2

are being carried out under the direction of Major Mark L. Ireland, of the Quartermasters' Corps, as a part of the extensive program of the advisory board of highway research of the National Research Council. It is expected that the tractive resistance of these trucks, as determined on the road, will soon be available for comparison with the laboratory dynamometer tests of the same trucks.

The trucks have been chosen as examples of the widest variation in internal friction. Yet the total tractive resistance, expressed in pounds per M, lies between the limits of 19 and 26. The higher values are clearly the fault of excessive friction in the transmission system, due probably to newness of the vehicle; hence they are likely to decrease with use.

These tests tend to prove that the rolling resistance of heavy trucks in good working order might be expressed as 20 to 21 lb. per M for all sizes. This figure differs from that of the group of 50 cars referred to in the previous paragraph and is expressed in equation (1). It agrees better with the group of seven cars mentioned in the next paragraph. These cars were tested recently and were known to be in free-rolling condition. Their rolling resistance was expressed by the figure, 19 lb. per M. This fairly close agreement among widely differing vehicles may lead to the further generalization that the rolling resistance of all rubber-tired vehicles, pneumatic and solid, when in best running condition, lies between the narrow limits of 19 and 21 lb. per M.

The lack of agreement with the 50 cars tested from 1916 to 1921 has been noted. This may be explained partly by the fact that most of the lighter cars were equipped with fabric tires, while the heavy cars were equipped with cord tires. Equation (1) yields 27 lb. per M for light cars but, on the other hand, it gives 19 lb. per M for a car weighing 4500 lb. It seems likely that with the increasing use of cord tires future tests will show less resistance for the lighter cars, thus bringing equation (1) nearer to the simpler value of 19 to 21 lb. per M for all vehicles.

FRICTION OF PARTS

Results have been given for the total chassis friction and its relation to the weight of the vehicle. It is now proposed to subdivide the total friction into several components to observe their relative importance better.

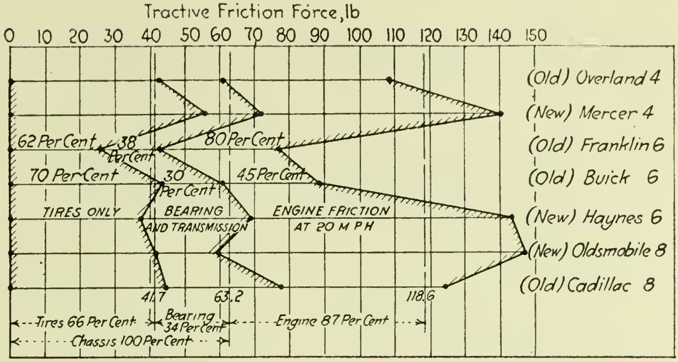

Fig. 6—Internal Friction of Automobiles

Five friction items have been chosen as giving the most useful information: (*a*) the front tires; (*b*) the front bearings; (*c*) the rear tires; (*d*) the rear bearings, including the transmission in the neutral gear; and (*e*) the engine friction in the direct drive when the rear wheels are turning at the equivalent of a car speed of 20 m.p.h.

Speed has not been mentioned in the first four items inasmuch as the friction is nearly constant at all speeds. Engine friction has been measured by driving through the rear wheels from the dynamometer, with the ignition shut-off. Care was taken to have the engine warm before beginning the measurements; also to have the throttle wide-open, as this was found to reduce the resistance materially. This last fact suggests that engine resistance is due in part to compression action in the cylinders or to air friction in the valves. It was observed that the engine resistance, measured in terms of tractive force at the rear wheels, increased directly as the speed. The frictional resistance of the engine at 40 m.p.h. was *double* that at 20 m.p.h. This result is significant by contrast with other friction measurements including those of tires, bearings and transmission, of which the increase of friction with speed is negligible. Granting that the engine friction determined in this way is too large, it may at least be used for comparative purposes.

Seven representative automobiles were chosen for the friction measurements. Table 2 gives the dimensions, weight, and the like, of the several cars, together with the friction items already mentioned. Considerable vari-

TABLE 2—FRICTIONAL DISTRIBUTION IN THE CHASSIS

Date of Test	April 3, 1922	April 4, 1922	April 10, 1922	April 11, 1922	April 11, 1922	April 12, 1922	April 12, 1922
Name of Car	Overland	Mercer	Buick	Cadillac	Oldsmobile	Franklin	Haynes
Car Model	85	......	44	......	47 FS	9A	55
Number of Cylinders	4	4	6	8	8	6	6
Engine Bore and Stroke, in.	4⅛x4½	3¾x6¾	3¼x4½	3⅛x5⅛	2⅞x4½	3¼x4	3½x5
Weight: Front Wheels, lb.	1,360	1,850	1,350	2,010	1,495	1,250	1,585
Rear Wheels, lb.	1,650	2,250	1,460	2,400	1,735	1,450	1,815
Total, lb.	3,010	4,100	2,810	4,410	3,230	2,700	3,400
Front Tires: Size, in.	32x4	32x4½	32x4	35x5	32x4	32x4	33x4
Kind	1 Fabric 1 Cord	Goodyear Cord	1 Fabric 1 Cord	Revere Cord	Federal Cord	Goodyear Cord	Goodyear Cord
Rear Tires: Size, in.	32x4	32x4½	32x4	35x5	32x4	32x4	33x4
Kind	Cord	K-S Cord	1 Fabric 1 Cord	Yale Cord	Federal Cord	Goodyear Cord	Goodyear Cord
Rolling Resistance:							
Front Tires, lb.	23.2	25.3	22.8	21.0	16.4	9.5	14.6
Front Bearings, lb.	4.4	2.4	3.2	6.2	4.0	7.4	3.9
Total Front Wheel, lb.	27.6	27.7	26.0	27.2	20.4	16.9	18.5
Rear Tires, lb.	19.0	30.0	20.3	23.5	25.2	16.0	22.8
Rear Bearings and Transmission, lb.	14.0	14.0	14.6	26.5	13.8	10.0	27.8
Total Rear Wheel, lb.	33.0	44.0	34.9	50.0	39.0	26.0	50.6
Total, Front and Rear Wheels, lb	60.6	71.7	60.9	77.2	59.4	42.9	69.1
Tires Only, lb	42.2	55.3	43.1	44.5	41.6	25.5	37.4
Bearings and Transmission, lb.	18.4	16.4	17.8	32.7	17.8	17.4	31.7
Engine, at 20 M.P.H., lb.	47.5	69.0	27.4	47.5	88.0	34.0	74.4
Total, Including Engine, lb.	108.1	140.7	88.3	124.7	147.4	76.9	143.5

ation was found in the different cars, as might have been expected from the difference in weight. To bring this out more clearly, the total friction, divided into the three parts of tires, bearings and transmission, and the engine, has been plotted in Fig. 6.

A study of the friction diagram shows that, in spite of variations of other factors, the ratio of tire friction to bearing-transmission friction remains constant in the order of 2 to 1. In other words, the friction loss in the tires is two-thirds the total chassis friction, exclusive of the engine.

Examination of the internal-friction chart reproduced in Fig. 6 shows that the item of engine friction is large, being on the average seven-eighths the remaining chassis friction. Admitting that the measured engine frictions are too large, for reasons already mentioned, it is probably true that comparisons can be made fairly between them. Three of the engines had been used but little since leaving the factory and showed higher friction than the others. Two had run over 12,000 miles since the last overhauling and were in free-running condition. Comparing the two groups, the new engines had nearly three times the friction of the old ones.

A true comparison of the frictional resistance of the chassis requires that all values be reduced to the same car-weight. Table 3 is made out for the seven cars, giving the resistance of the tires, bearings and transmission, and chassis without the engine, in terms of pounds

TABLE 3—INTERNAL FRICTION AND TOTAL FRICTION, IN POUNDS OF TRACTIVE FORCE PER 1000 LB. OF WEIGHT

Name of Car	Weight, Lb.	Internal Friction, Lb.		Total Friction, Lb.
		Tires	Bearings and Transmission	
Buick.......	2,810	15.4	6.3	21.7
Cadillac.....	4,410	10.1	7.5	17.6
Franklin.....	2,700	9.5	6.4	15.9
Haynes......	3,400	11.0	9.3	20.3
Mercer......	4,100	13.6	4.0	17.6
Oldsmobile...	3,230	13.0	5.4	18.4
Overland....	3,010	14.0	6.0	20.0
Average...	3,380	12.4	6.4	18.8

per thousand pounds. The items for the different cars are by no means constant, yet they are not widely apart. The average of the three columns can be considered as fairly representative of the friction items for the cars weighing from 3000 to 5000 lb. From these results the following conclusions can be drawn: In cars weighing from 2500 to 5000 lb. the average friction of cord tires only was 12.5 lb. per M; of bearings and transmission only, 6.5 lb. per M; and of total chassis without the engine, 19 lb. per M.

RESISTANCE FORMULAS

Two rules for determining the total chassis friction have been deduced from the dynamometer tests. Their agreement logically should be compared. The first was deduced from 50 tests of various cars where the front and the rear resistances were measured separately. This embraced a variety of tire equipment that was mostly fabric tires on the lighter cars and cord tires on the heavier ones. These tests were spread over a period of 5 years. The formula for the first series, where L = the weight of the chassis, is

$$\text{Total chassis resistance} = 0.012\,L + 30$$

The second series consisted of seven cars, where the tire and the bearing resistances were separately measured. Practically all the cars were equipped with cord tires and the measurements were made at one time by the same observers. The formula for the second series is

$$\text{Total chassis resistance} = 0.019\,L$$

The divergence of the two formulas can be seen best by comparing the values for the same weights of car. For $L = 2500$, the resistance by the two formulas is: first, 60 lb.; second, 47.5 lb.; difference, 12.5 lb. For $L = 4000$, the two formulas give first, 78 lb.; second, 76 lb.; difference, 2 lb. From this comparison it appears that the first formula gives larger values for the light-weight cars, while both agree well for the heavier cars. The theory has been advanced that the general use of fabric tires on the lighter cars was the cause of their increased resistance.

The internal friction of several heavy trucks has been separated into tire and bearing elements as shown in Table 4.

In Table 4, Truck B 432,788 was known to have the excessive transmission friction incident to a new truck, which is shown by its bearing and transmission friction of 12.0 lb. per M, or nearly double that of its companion truck. The other two trucks can be considered as representatives of the heavy class. Their average tire friction is 13.5 lb. per M, or 61 per cent; average bearing and transmission friction, 8.5 lb. per M, or 39 per cent; total, 22.0 lb. per M, or 100 per cent.

The corresponding figures for passenger cars with cord tires were: average tire friction, 12.5 lb. per M, or 66 per cent; average bearing and transmission friction, 6.5 lb. per M, or 34 per cent; total, 19.0 lb. per M, or 100 per cent. In general, the friction distribution in trucks does not differ greatly from that in lighter vehicles.

TABLE 4—INTERNAL FRICTION OF HEAVY TRUCKS

Name of Truck	Tractive Resistance in Percentage and Pounds per 1000 Lb. of Weight		
	Tires Only	Bearings and Transmission	Total
Quartermaster Standard, Type B, No. 432,799	13.7 lb. 67%	6.6 lb. 33%	20.3 lb. 100%
Quartermaster Standard, Type B, No. 432,788	15.1 lb. 56%	12.0 lb. 44%	27.1 lb. 100%
Mack 7½-Ton, Type-AC, Chain Drive	13.4 lb. 56%	10.4 lb. 44%	23.8 lb. 100%

Fabric Tires

The influence of load and inflation-pressure on the rolling resistance of pneumatic tires becomes evident when the results of tests are studied. Some typical examples will now be presented for a line of fabric tires from the same maker, sizes 32 x 4 in., 33 x 4½ in. and 35 x 5 in. In making tests of the rolling resistance of these tires, an arbitrary schedule of loads and inflation-pressures was adopted, designed to cover a sufficiently wide range for the size of the tires. Three different speeds were used for each test-load and inflation-pressure, resulting in as many as 36 independent readings of rolling resistance for one tire. The changes due to speed were very slight; hence it was possible to eliminate the speed as a variable, giving an average value for each load and inflation-pressure without regard to the speed. The rolling resistance of each tire, arranged in columns under the respective loads, is given in Table 5. The variation of the figures in each column is a measure of the effect produced by the change of inflation-pressure.

To bring out the variation more clearly, a parallel column has been added to Table 5 giving the results in percentages, using 100 per cent for the smallest value of the resistance. The increase of rolling resistance due to lower inflation-pressure is apparent in every case. A general conclusion from the figures in Table 5 can be stated thus: The rolling resistance of a fabric tire, fully loaded and inflated, may be increased by more than 50 per cent when the inflation-pressure is dropped from 90 to 30 lb. per sq. in., and by 25 per cent when the pressure is dropped to 50 lb. per sq. in. A word of caution is added in this connection, namely, that the total car re-

TABLE 5—ROLLING RESISTANCE OF FABRIC NON-SKID TIRES AT 20 TO 40 M.P.H. SPEEDS

Test No. 28; Size, 32x4 In.; Weight of Tire, 19.5 Lb.; Outside Diameter 32.8 In.

Inflation-Pressure, Lb. per Sq. In.	Rolling Resistance		Rolling Resistance		Rolling Resistance	
	Load, 460 Lb.		Load, 700 Lb.		Load, 975 Lb.	
	Lb.	Per Cent	Lb.	Per Cent	Lb.	Per Cent
30	8.03	186	14.45	165	22.3	163
55	5.35	124	10.60	121	16.8	121
80	4.33	100	8.78	100	13.7	100

Test No. 60; Size, 33x4½ In.; Weight of Tire, 23.5 Lb.; Outside Diameter, 33.7 In.

	Load, 585 Lb.		Load, 935 Lb.		Load, 1,285 Lb.	
30	9.05	124	18.40	152	27.7	145
45	8.65	118	15.60	130	23.0	121
65	8.65	118	14.20	118	21.2	111
90	7.32	100	12.03	100	19.1	100

Test No. 102; Size, 35x5 In.; Weight of Tire, 33.5 Lb.; Outside Diameter, 36.0 In.

	Load, 750 Lb.		Load, 1,050 Lb.		Load, 1,650 Lb.	
40	12.70	123	19.80	128	37.0	148
55	11.40	109	17.60	115	32.4	129
70	10.50	102	15.90	103	28.2	113
90	10.30	100	15.40	100	25.1	100

sistance will not be increased in this same ratio, because the tires produce only a part of the total friction of the chassis; also, that this result applies only to *fabric* tires.

The diagrams shown in Figs. 7, 8 and 9 contain a curve for each inflation-pressure, plotted with the rolling resistance and the load as coordinates. The plotted points show the existence of slight observational errors, yet fairly satisfactory curves can be drawn through them. An interesting fact, shown on all the diagrams, is that the line for all the inflation-pressure curves is straight. Another is that, at the same inflation-pressure, these straight lines practically coincide on each diagram, showing that the rolling resistance depends solely upon the load and not upon the size of the tire. Whether this conclusion can be extended to the larger sizes, such as pneumatic tires for trucks, will be discussed elsewhere.

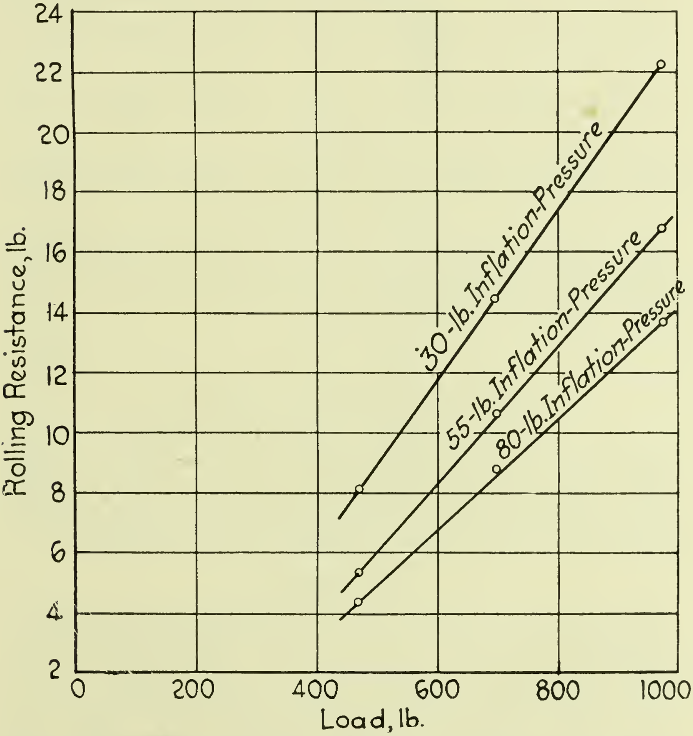

Fig. 7—Average Rolling Resistance of a 32 x 4-In. Fabric Pneumatic Tire at Speeds of from 20 to 40 M.P.H. and Different Inflation-Pressures

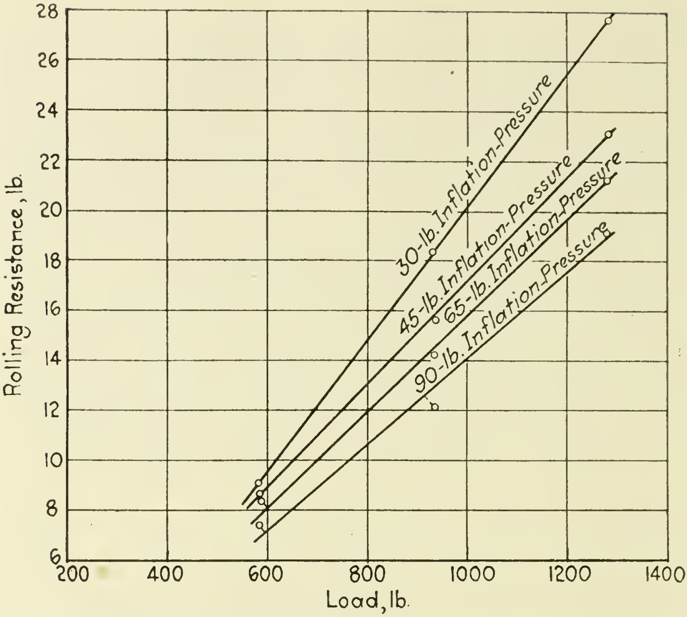

FIG. 8—AVERAGE ROLLING RESISTANCE OF A 33 X 4½-IN. FABRIC PNEUMATIC TIRE AT SPEEDS OF FROM 20 TO 40 M.P.H. AND DIFFERENT INFLATION-PRESSURES

The rolling resistance of the fabric tires, when fully inflated, can be expressed by a simple algebraic formula which applies to all three sizes:

$$R = 0.018\,L - 3.0 \qquad (2)$$

where,

L = the load on the tire, in pounds
R = the rolling resistance, in pounds

Another way of expressing the result is in the simple form of pounds per thousand pounds. This number varies slightly with the load. Its average value is 15 lb. per M for the tire sizes referred to in this article.

CORD TIRES

Dynamometer tests of cord tires show less rolling resistance than those of fabric tires. The difference can be stated as being approximately one-third. In other words, the rolling resistance of a standard make of cord tire is only two-thirds that of a fabric tire. This conclusion has been proved many times during the last 5 years with different sizes and makes of tire. The figures

TABLE 6—COMPARATIVE ROLLING FRICTION OF FABRIC AND CORD TIRES

Size of Tire, in.	32x4	33x4½	35x5
Speed, m.p.h.	20 to 40	20 to 40	20 to 40
Inflation-Pressure, lb. per sq. in.	30 to 80	30 to 90	40 to 90
Load, lb.	460 to 975	585 to 1,285	750 to 1,650
Number of Readings	54	72	32
Average of Rolling-Friction Readings			
Fabric, lb.	11.30	15.20	21.10
Cord, lb.	7.60	9.06	13.45
Fabric, per cent.	100.00	100.00	100.00
Cord, per cent.	67.00	60.00	64.00

in Table 6, based on tests of Fisk and Goodyear tires, both of which have shown practically identical rolling resistances, are presented as typical.

The deductions from these figures are confirmed by

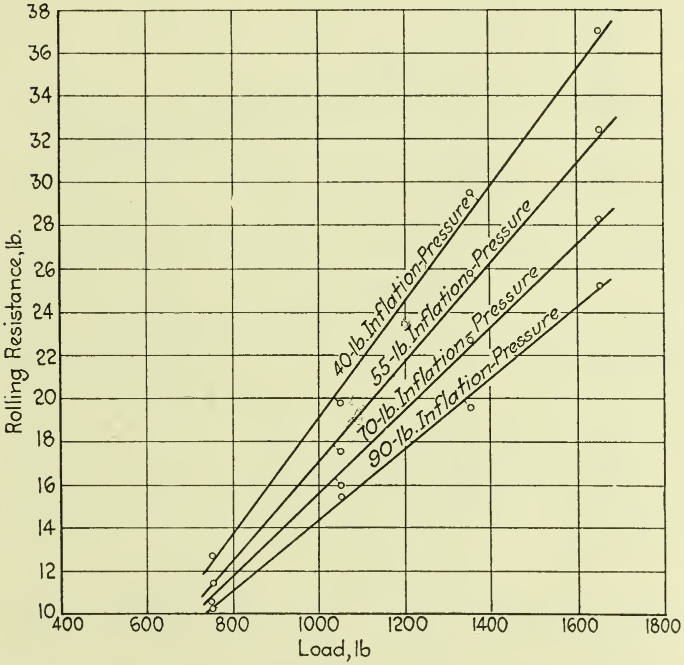

FIG. 9—AVERAGE ROLLING RESISTANCE OF A 35 x 5-IN. FABRIC PNEUMATIC TIRE AT A SPEED OF 20 M.P.H. AND DIFFERENT INFLATION-PRESSURES

curves plotted from the test results. An example is given in Fig. 10, where two rolling-resistance curves are plotted, one for fabric and one for cord tires. The curves are nearly parallel, showing that the difference in friction is about the same at all inflation-pressures.

The cord tire has a valuable characteristic that is clearly shown in Fig. 10. A low friction is reached at a moderate inflation-pressure. In this case the resistance is 11.5 lb. at an inflation-pressure of 65 lb. per sq. in., and 11.0 lb. at a pressure of 90 lb. per sq. in.; hence, the cord tire can be run at a moderate inflation-pressure with only a slight loss of power. On the other hand, the fabric-tire resistance diminishes steadily as the inflation-

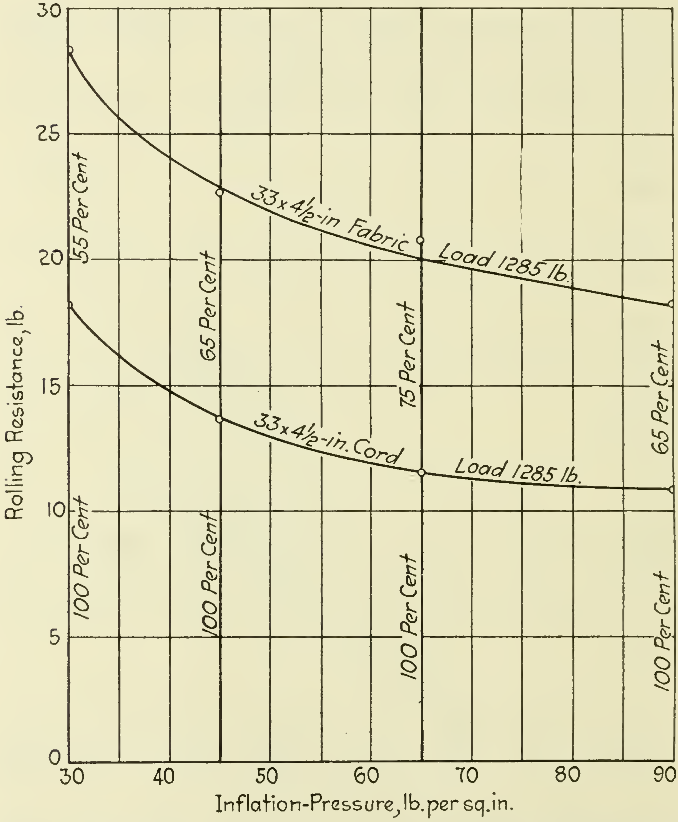

Fig. 10—Typical Rolling-Resistance Curves of Fabric and Cord Tires at a Constant Load and Varying Inflation-Pressure

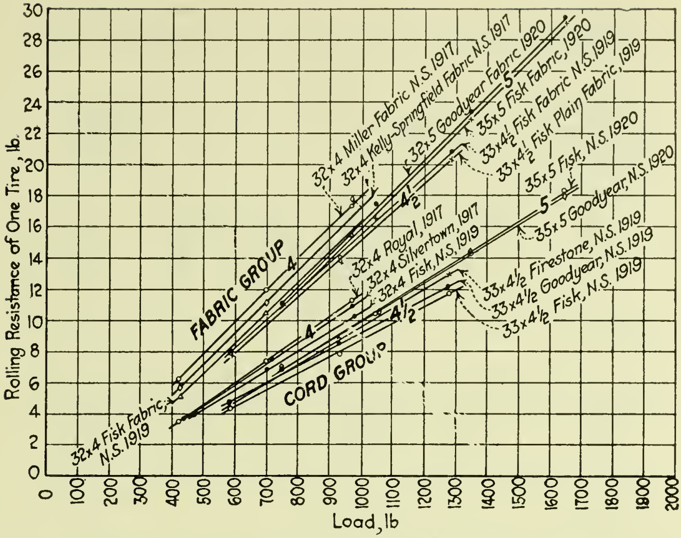

Fig. 11—Comparative Rolling-Resistance Curves of Fabric and Cord Tires at an Inflation-Pressure of 65 Lb.

pressure is increased; hence, high pressures are demanded for the saving of power.

Another illustration of the difference between fabric and cord-tire resistances is given in Fig. 11. In this example the curves of rolling resistance are plotted for a constant inflation-pressure. The results are given for several sizes of fabric and cord tires. There are differences between the various sizes, but the points lie in a band, or strip, in each case. These curves show clearly that the rolling resistance of any size of cord tire, under any load, is about two-thirds that of a fabric tire under the same conditions.

The rolling resistance of cord tires is further analyzed in Table 7, which gives the rolling friction of 33 x 4-in. tires arranged for three speeds, three loads and four inflation-pressures, a total of 36 readings. The figures in Table 7 are compiled from tests of seven cord tires of well-known makes, and therefore can be considered as fairly representative of new cord tires.

The tabular values have been plotted in the diagram, Fig. 12, with one curve for each 10 lb. of inflation-pressure. The spacing of the curves shows clearly the change of resistance with any change of pressure. After reaching an inflation-pressure of 60 lb. per sq. in., but

TABLE 7—AVERAGE ROLLING FRICTION FOR SEVEN RIBBED AND NON-SKID FIRESTONE, FISK, GOODRICH AND GOODYEAR TIRES

Inflation-Pressure, Lb. per Sq. In.	Load, Lb.	Speed, M.P.H.			Average Rolling-Friction, Lb.
		20	30	40	
30	585	6.03	6.33	6.91	6.42
45		5.16	4.98	5.26	5.13
65		4.68	4.70	4.65	4.68
90		4.17	4.29	4.54	4.33
Average					5.14
30	935	10.99	11.75	12.37	11.70
45		9.04	9.76	10.01	9.60
65		8.21	8.34	8.43	8.33
90		7.49	8.01	8.22	7.90
Average					9.38
30	1,285	17.39	18.17	19.01	18.19
45		14.06	14.71	14.89	14.55
65		12.16	12.32	12.64	12.37
90		10.87	11.30	11.70	11.29
Average					14.10
General Average					9.54

little further decrease in rolling friction is found, again showing that in cord tires a high degree of inflation is not required for easy rolling.

The resistance curves are practically straight lines of varying slope. An empirical formula has been derived for these curves that gives the rolling resistance for any load and inflation-pressure within the limits covered by the experimental work

$$R = [(L - 200) \div 100] \times \{ [1.06 + (90 - P)^2] \div 7000 \} \qquad (3)$$

where

$L =$ Load
$P =$ Inflation-pressure
$R =$ Rolling resistance

The limits of load are from 500 to 1300 lb., and those of inflation pressure, 30 to 90 lb. per sq. in. This formula

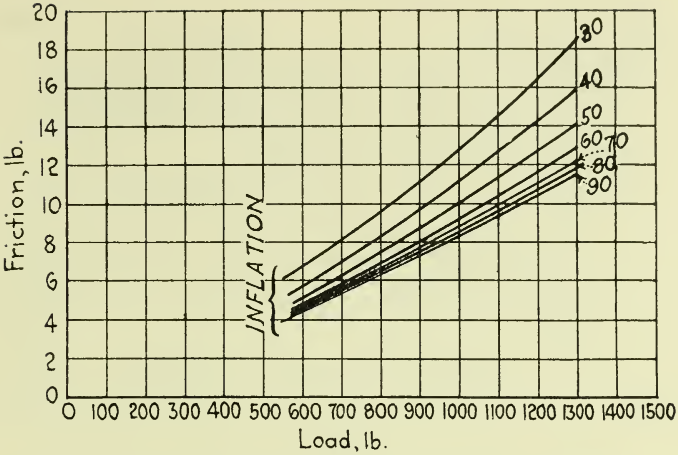

FIG. 12—ROLLING FRICTION OF 33 x 4½-IN. CORD TIRES AT DIFFERENT LOADS AND INFLATION-PRESSURES RANGING FROM 30 TO 80 LB.

can be put into simpler form by inserting a normal inflation-pressure of 60 lb. per sq. in., causing it to take the form

$$R = 0.012\,L - 2 \qquad (4)$$

A similar formula was deduced for fabric tires, where the constants are exactly 50 per cent larger than for the cord-tire formula. This is further evidence that cord tires have two-thirds the rolling friction of fabric tires.

SOLID-RUBBER TIRES

Tests of solid-rubber tires were made at intervals from 1919 to 1921 on ¾ to 2-ton trucks. It was found that solid-tire resistances could be measured by the jacking-up method and that, in general, the resistance was between those of cord and fabric tires under the same load.

In 1921, in cooperation with the Committee on the Tractive Resistance of Roads Research, the drums were widened to accommodate the dual tires of heavy trucks. Six trucks were brought to the Mason Laboratory for the investigation of internal friction by the dynamometer method. As planned by Major Ireland, the tests were carried out with empty and with loaded trucks, with different temperatures of the lubricant, with several kinds of new solid-rubber tires, and with a few well-worn tires. The tire-resistance measurements are summarized as shown in Table 8.

The following conclusions are apparent from the figures in Table 8:

(1) Partly worn tires roll with less friction than new and more resilient tires, the difference being from 20 to 25 per cent; hence, increased cushioning must be paid for by increased power

(2) Front tires show more friction than rear tires, both when new and when worn. This is to be explained by the relatively heavier load carried by the front tires. The rear-tire tests were made with the truck empty; hence, the load per inch of tire width was considerably less than that for the front wheels

(3) The rolling resistance of the solid-rubber tires varies considerably, depending upon the wear, the

TABLE 8—SUMMARY OF TIRE-RESISTANCE MEASUREMENTS

		Tire Size, In.	Load per Inch of Tire Width, Lb.	Tire Resistance per 1,000 Lb. of Weight, Lb.
New Front Tires	Single	36x4	583	21.7
		36x5	466	19.0
		36x6	375	20.2
		36x7	375	19.0
		Average..	450	20.0
Worn Front Tires	Single	36x5	466	15.5
		36x6	375	16.5
		36x7	375	16.2
		Average..	400	16.1
New Rear Tires	Dual	40x6	275	18.0
		40x6	275	15.7
	Single	40x12	275	14.0
		40x12	275	15.8
		40x12	275	19.0
		Average..	275	16.5
Worn Rear Tires	Dual	40x6	285	12.5
		40x6	255	11.0
		Average..	270	11.8

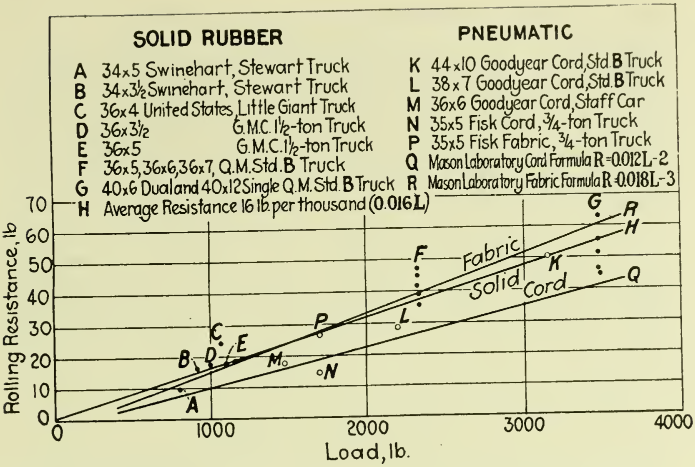

FIG. 13—ROLLING RESISTANCE OF SOLID-RUBBER AND PNEUMATIC TIRES FOR DIFFERENT LOADS

construction and the like, but the values are all between the cord and the fabric-tire figures. Taking the average of all the readings as an index, the solid-tire resistance is 16.0 lb. per M

A final comparison of solid and pneumatic tires has been made in Fig. 13. All the solid-rubber tire resistances have been plotted with respect to the load in three groups; for light trucks, and for the front and rear wheels of heavy trucks. The average of all the solid-rubber-tire results has been represented by the line marked 16 lb. per M, as mentioned above. The results for several large-size pneumatic-tires also are plotted in Fig. 13 and the Mason Laboratory formulas for fabric and cord tires are stated. Fig. 13 confirms the previous statement that solid-rubber-tire friction is approximately equal to fabric-tire friction.

Dynamometer tire-friction measurements have been made at speeds of 20, 30 and 40 m.p.h. Over this range the observed readings have been nearly constant. Stated more correctly, the variations due to speed have been of about the same order as the observational error of a single reading; hence they are not easily observed. When the averages of many different readings are taken, a slight increase of the rolling resistance with the speed is found. The figures in Table 9 show that, by doubling the speed, the rolling resistance may be increased by 6 or 8 per cent.

In view of the small change produced by the speed, it has been deemed most satisfactory to ignore it and to give the average value as constant for all speeds. This assumption is allowable for rolling on smooth, hard surfaces. For rough, uneven or soft roads, the rolling resistance doubtless will increase materially with the speed.

Careful measurements have shown that the non-skid type of tread produces slightly more resistance than the smooth tread. The increase produced by the non-skid tread is small and is presented as a matter of interest rather than importance. The figures in Table 10, based on tests of three well-known tires, will indicate the influence of the non-skid tread.

TABLE 9—AVERAGE ROLLING RESISTANCE AT DIFFERENT SPEEDS FOR 11 33x4½-IN. TIRES

Speed, M.P.H.	Series No. 1		Series No. 2	
	Inflation-Pressure, 65 Lb. per Sq. In.		Inflation-Pressure, 90 Lb. per Sq. In.	
	Load, 935 Lb.		Load, 1,285 Lb.	
	Average Resistance		Average Resistance	
	Lb.	Per Cent	Lb.	Per Cent
20	9.4	100	12.6	100
30	9.8	104	13.1	104
40	9.9	106	13.6	108

The figures in Table 10 seem to show that the effect of the non-skid tread is to increase the rolling resistance about 5 per cent. The increase is caused, probably, by the greater compression of the rubber under the load and may vary somewhat with the shape of the non-skid markings.

The influence of heavy-wall tubes of the "non-puncture" type has been measured by running alternate tests with thin-wall and thick-wall tubes in the same casing. The results in two instances have shown slightly greater friction for the thick-wall tubes, the difference being about 5 to 10 per cent.

TABLE 10—INFLUENCE OF NON-SKID TREAD ON ROLLING RESISTANCE[3]

Kind of Tire, Cord	Rolling Resistance	
	Ribbed Tread, Lb.	Non-Skid, Tread, Lb.
Fisk	8.86	9.23
Firestone	8.84	9.52
Goodyear	8.80	9.98
Average	8.83	9.58

[3] Average of 36 readings on 33 x 4½-in. cord tires at different speeds, loads and inflation-pressures. The increase in resistance due to the non-skid tread is 8 per cent.

Similar results have been obtained from a tube protector, consisting of a thick endless pad lying between the tube and the casing. The protecting strip was found to increase the rolling resistance by 7 per cent, in a 33 x 4½-in. cord-tire casing.

LARGE PNEUMATIC TIRES

Tests of the smaller sizes, 4 and 5-in. tires, have shown that the rolling resistance is practically identical for the same loads and inflation-pressure. If this holds true, it follows that a single curve or formula for rolling resistance would hold for all sizes of a given type, such as a cord tire of normal inflation-pressure. Unfortunately for this theory, tests of the larger sizes have shown somewhat higher friction, proportionately, than those of the smaller sizes. The values in Table 11 are averages for several sizes of cord tire as measured on the drums.

TABLE 11—ROLLING RESISTANCE OF LARGE PNEUMATIC TIRES

Tire Size, In.	Rolling Resistance per 1,000 Lb. of Weight, Lb.
32x4	11.0
33x4½	11.0
35x5	11.0
36x6	12.5
38x7	13.0
44x10	16.0

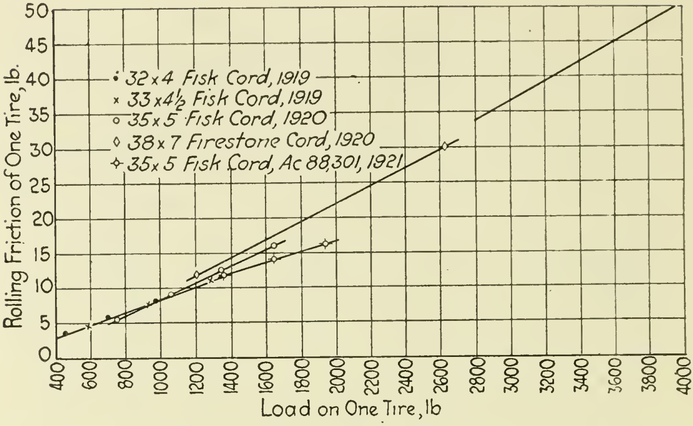

FIG. 14—ROLLING RESISTANCE OF DIFFERENT SIZES OF CORD TIRES AT AN INFLATION-PRESSURE OF 90 LB.

Fig. 14 affords a comparison of several sizes of cord tire under different loads. It is probably true that the larger sizes were less well inflated than the smaller sizes. In one case, that of the 44 x 10-in. size, the tire was marked with the direction to "Inflate to 140." The test was made at a pressure of 100 lb. per sq. in. because of a limitation of the available air-supply at that time. It can be concluded that the larger-sizes of pneumatic tire will have somewhat higher rolling resistance, proportionately, than the smaller sizes. This statement can be qualified by the further assumption that if all the sizes were kept at the inflation-pressures recommended by the manufacturers, the rolling resistances would be practically alike.

The evidence of the effect of wear on the rolling resistance of pneumatic tires is rather conflicting, since some old tires have shown excessive friction, while others have agreed well with that of new tires. Close inspection proves that in a majority of cases the rolling frictions of a new and of an old tire are about the same; also that in exceptional instances some reason for the difference usually can be found as when old tires get out of shape and run untrue on the rims. Reliable information on the effect of age and wear on tires is difficult to obtain but fortunately this is not a matter of practical importance. From the evidence at hand it is believed that a worn pneumatic tire has about the same rolling friction as a new tire of the same kind. Solid-rubber-

TABLE 12—ROLLING RESISTANCE OF 32 X 4-IN. PNEUMATIC TIRES

Name	Number	Description	Year	Inflation-Pressure, Lb. per Sq. In	Load, Lb.		
					500	700	1,000
Miller	3	Fabric, non-skid	1917	30	8.8	14.0	21.5
				55	7.2	11.7	18.5
				80	6.5	11.0	17.5
Kelly-Springfield	12	Fabric, non-skid	1917	30	9.1	15.5	25.0
				55	7.7	13.1	21.2
				80	6.3	10.5	17.1
Fisk	27	Fabric, plain	1919	30	7.7	12.3	20.0
		Weight, lb. 18.0		55	6.5	10.5	16.5
		Diameter, in. 32.6		80	4.9	8.0	13.0
Fisk	28	Fabric, non-skid	1919	30	8.8	14.4	22.8
		Weight, lb. 19.5		55	6.1	10.5	17.4
		Diameter, in. 33.0		80	5.0	8.6	14.0
Firestone	16	Cord, non-skid	1917	30	8.4	12.5	18.9
				55	6.1	9.3	14.0
				80	5.5	8.5	13.3
Goodrich	14	Cord, ribbed	1917	30	6.5	10.5	17.2
				55	5.5	8.0	12.5
				80	4.8	6.8	10.2
Goodyear	15	Cord, non-skid	1917	30	7.8	12.3	19.5
				55	6.0	8.9	13.6
				80	5.0	7.4	11.3
Miller	6	Cord, non-skid	1917	30	8.2	12.1	19.5
				55	5.2	8.0	14.0
				80	4.5	7.0	11.2
U. S. Royal	8	Cord, non-skid	1917	30	8.0	12.1	19.8
				55	4.9	7.5	12.7
				80	3.9	5.9	10.8
Fisk	26	Cord, ribbed	1919	30	5.9	8.6	15.0
		Weight, lb. 21.0		55	4.5	6.5	10.6
		Diameter, in. 33.3		88	3.8	5.5	8.5
Fisk	25	Cord, non-skid	1919	30	5.7	9.0	14.0
		Weight, lb. 21.5		55	4.7	7.2	11.4
		Diameter, in. 33.3		80	4.0	6.0	10.3
Fisk	29	Cord, non-skid	1922	30	7.2	9.5	15.0
		Weight, lb. 22.0		55	6.3	8.0	11.7
		Diameter, in. 33.3		80	5.5	7.3	10.8
Fisk	30	Cord, non-skid	1922	30	6.5	9.5	16.3
		Weight, lb. 22.2		55	5.0	7.6	12.5
		Diameter, in. 33.3		80	4.6	6.8	11.3
Yale	31	Cord, non-skid	1922	30	9.0	12.5	22.0
		Weight, lb. 25.0		55	7.0	9.5	15.0
		Diameter, in. 33.75		80	6.5	8.5	13.0

tire resistance is known to decrease as the tire wears out, leaving a thinner layer of rubber.

The Mason Laboratory tests give evidence that there has been a general decrease in the rolling resistance of pneumatic tires for several years. Tests of tires of the same size have been carried on under the same conditions of loading and inflation; hence the results should be comparable. In the 32 x 4-in. size, a considerable decrease was observed from 1917 to 1919 in both fabric and cord tires as shown in Table 12. This has been partly lost between 1919 and 1922. The decrease in rolling friction from 1917 to 1919 is shown in detail in several of the diagrams of comparative results.

The 35 x 5-in. size referred to in Table 13 was first tested in the Mason Laboratory in 1920. Fabric and cord tires of several different makes were compared and showed a satisfactory agreement in rolling friction. Tests of cord tires of this size have shown a constant decrease of friction, amounting to about 10 per cent.

TABLE 13—ROLLING RESISTANCE OF 35 X 5-IN. PNEUMATIC TIRES

Name	Number	Description	Year	Inflation-Pressure, Lb per Sq. In	Load, Lb.			
					700	1,000	1,300	1,600
Goodyear	100	Fabric, non-skid	1920	40	12.8	20.0	27.5	35.0
		Weight, lb. 31.0		55	11.2	17.5	24.1	30.5
		Diameter, in. 35.9		70	10.0	15.8	21.7	27.7
				90	9.5	15.0	20.6	26.2
Fisk	102	Fabric, non-skid	1920	40	11.5	19.6	27.8	36.0
		Weight, lb. 33.5		55	10.5	17.4	24.5	31.5
		Diameter, in. 36.0		70	9.7	15.6	21.5	27.3
				90	9.4	14.4	19.3	24.0
Yale	104	Fabric, non-skid	1920	40	14.0	22.8	31.5	40.5
		Weight, lb. 36.25		55	12.7	19.5	26.2	33.0
		Diameter, in. 36.0		70	11.7	17.5	23.3	29.2
				90	10.7	15.5	20.4	25.2
Goodyear	101	Cord, non-skid	1920	40	9.0	13.5	18.5	23.2
				55	7.3	11.2	15.3	18.3
				70	6.0	9.6	13.5	17.3
				90	5.0	8.5	12.4	16.0
Fisk	103	Cord, non-skid	1920	40	8.5	12.8	18.2	24.0
		Weight, lb. 35.5		55	6.8	11.1	15.5	20.0
		Diameter, in. 37.2		70	5.9	9.6	13.4	17.2
				90	4.8	8.5	12.0	15.5
Fisk	109	Cord, non-skid special	1921	40	......	11.5	15.5	20.0
		Weight, lb. 37.0		55	......	10.3	13.6	17.0
		Diameter, in. 37.2		70	......	9.0	12.0	14.7
				90	......	8.3	11.0	13.5

The 33 x 4½-in. tires of 1919 cited in Table 14 were characterized by a low rolling-friction as shown by Fig. 12. This feature seems to have been lost since that time, and recent tests show that a tire of this size now has more rolling-friction than a 32 x 4-in. size.

Comparisons of tire tests made several years apart require proof that the differences really exist in the tires and not in the apparatus itself. It can be said that, at least during the last 4 years, while the comparisons have been in progress, cord-tire friction appears to have decreased noticeably. It is noteworthy that tires made by several of the leading manufacturers have shown marked uniform rolling resistances. This indicates, possibly, that the processes of tire manufacture have become standardized.

Deviation from this uniformity has been observed in a few instances. In one case, a pair of tires marked "Cord" was found to have the resistance usually found in fabric tires, that is, 50 per cent in excess of cord tires. In two other cases, cord tires showed 27 and 30-per cent excess over the standard cord-tire figures. All the examples that deviated from the standard values were found in the products of the smaller and less well-known manufacturers.

TEMPERATURE AND HEAT

That rubber tires get warm while running is well known. The true source of the heat is now known to be within the tire structure. It may be due in part to compression of the rubber or in part to the flexure of the carcass or both. Heat generation is not confined to pneumatic tires but is found in the same degree in those of solid-rubber. This proves that compression of the rubber unaided by flexure of the tire walls will result in heat.

The flexure of a tire can be likened to the bending and unbending of annealed wire, where motion in any direction absorbs work and generates heat within the material. An ideal tire structure would be one of perfect elasticity, where the work of flexure or compression would be completely returned after the original positions were resumed. Present tire structures are at least partly elastic as indicated by the fact that cord tires have only two-thirds the rolling resistance of fabric tires; hence they have only two-thirds the heat-generating capacity.

It can be assumed fairly that the generation of heat and the resulting rise of temperature are directly associated with the rolling resistance of the tire. It can be assumed also that the amount of heat generated is exactly equivalent to the work of the rolling resistance and can be computed from this work, as in the case of the friction brake. This suggests a simple but practical way of testing the rolling resistance of two tires, which is to place them on the opposite wheels of a car and observe the surface temperatures after a long run. Measurements of the surface temperature are unsatisfactory as a final measure of tire resistance, although they may have some value as a check.

The surface temperature can be computed, approximately, from the rolling resistance. Useful conclusions can be drawn from such computations. It can be shown, for instance, that a large-size pneumatic-tire will heat more than a small one. The heat equivalent of the rolling resistance in British thermal units per hour can be found by obtaining the product of the resistance times the speed times 2546/375. The heat dissipated can be expressed as the product of the cooling surface or area times the difference of temperature between this surface and the surrounding air times a coefficient. The coefficient of heat transfer is fairly well known from experiments on moving air over surfaces, such as hot-blast heaters, automobile radiators and the like.

Applying this method to a 32 x 4-in. cord tire at a speed of 30 m.p.h., a computed rise of temperature of 29 deg. fahr. was obtained, using a normal full load of 1000 lb. Applying the method to a 44 x 10-in. cord tire at a speed of 30 m.p.h., with a load of 7000 lb., a computed rise of temperature of 90 deg. fahr. was obtained. This increase of temperature is due to the fact that the load capacity and the resultant heat of tire-friction increased faster than the surface for heat dissipation. The computed temperature-rise of small-size tires has been satisfactorily checked on the Mason Laboratory drums, but thus far no observations have been made on the large sizes.

Gearbox Lubricant

The transmission friction may be materially affected by the temperature of the lubricant in the gearbox. This is true especially of trucks, where the transmission friction normally is large. Tests were made by leaving the truck exposed all night to November weather, then run-

TABLE 14—ROLLING RESISTANCE OF 33 X 4½-IN. PNEUMATIC TIRES

Name	Number	Description	Year	Inflation-Pressure, Lb. per Sq. In	Load, Lb.		
					700	1,000	1,300
Fisk	59	Fabric, plain	1919	30	13.0	21.0	28.7
		Weight, lb. 20.0		45	10.5	16.5	22.7
		Diameter, in. 33.2		65	9.5	15.0	20.5
				90	7.7	12.5	17.0
Fisk	60	Fabric, non-skid	1919	30	12.0	20.0	28.0
		Weight, lb. 23.5		45	10.5	17.0	23.5
		Diameter, in. 33.7		65	9.5	15.5	21.5
				90	8.5	13.8	19.0
Firestone	55	Cord, ribbed	1919	30	7.5	12.3	17.0
		Weight, lb. 30½		45	6.0	9.6	13.3
		Diameter, in. 34.8		65	5.3	8.5	11.7
				90	5.0	8.0	11.0
Firestone	54	Cord, non-skid	1919	30	7.5	12.0	16.5
		Weight, lb. 30.5		45	7.0	11.0	15.0
		Diameter, in. 34.9		65	6.1	9.8	13.3
				90	5.5	8.7	11.9
Goodyear	68	Cord, ribbed	1919	30	7.6	12.8	18.0
		Weight, lb. 26.5		45	6.0	10.5	15.0
		Diameter, in. 34.1		65	4.8	8.0	11.5
				90	4.3	7.3	10.5
Goodyear	56	Cord, non-skid	1919	30	9.2	13.7	18.3
				45	7.3	11.0	14.7
				65	6.5	10.0	13.4
				90	5.7	9.2	12.5
Fisk	61	Cord, ribbed	1919	30	6.7	11.3	17.5
		Weight, lb. 24.0		45	6.2	9.5	14.0
		Diameter, in. 33.6		65	5.6	8.5	11.9
				90	5.3	8.0	10.9
Fisk	65	Cord, non-skid	1919	30	7.5	13.0	18.7
				45	6.3	10.3	14.0
				65	5.5	8.7	11.7
				90	5.3	8.5	11.4
Fisk	72	Cord, non-skid	1922	30	10.5	16.0	25.5
		Weight, lb. 27.0		45	8.8	13.0	19.2
		Diameter, in. 34.4		65	7.7	11.5	17.3
				90	7.3	10.7	16.0
Fisk	73	Cord, non-skid	1922	30	11.5	16.5	22.5
		Weight, lb. 29.5		45	9.0	13.4	18.0
		Diameter, in. 34.4		65	7.7	10.8	16.6
				90	6.5	9.5	14.8

ning the rear end on the drums and making friction measurements at intervals as the lubricant warmed-up. Runs were continued until the lubricant temperature had risen from 35 deg. fahr. to about 135 deg. fahr.

Fig. 15 shows the observed rise of temperature and the drop in the rear-end resistance of three trucks. In two cases the rear-end friction was increased 50 per cent by the coldest lubricant. The effect of friction is to produce heat that warms the gearbox and its contents, thus automatically reducing the friction. A temperature of 140 deg. fahr. in the gearbox was observed that was produced solely by internal friction. No attention was paid to the gearbox temperatures of passenger cars. All tests were made at the ordinary room-temperature with the engine and the transmission well warmed by preliminary running.

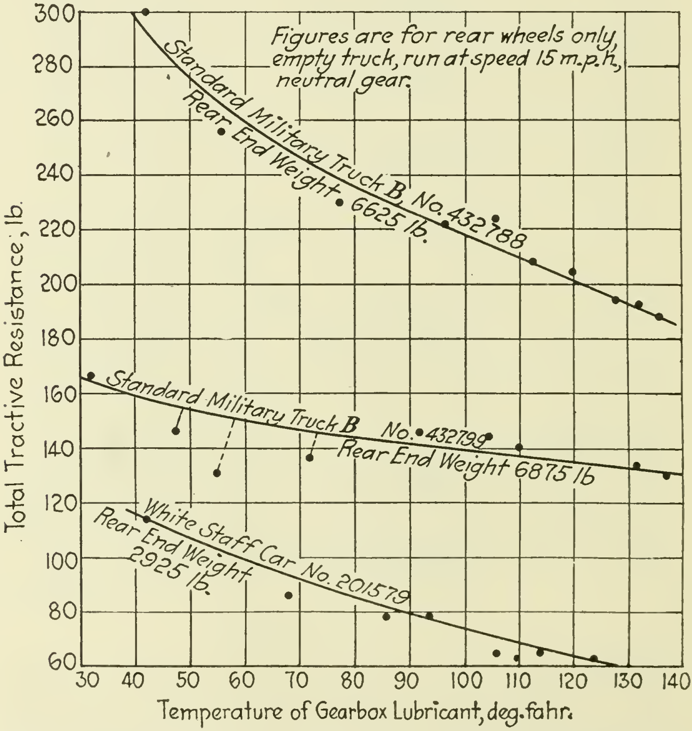

FIG. 15—CURVES SHOWING THE TRACTIVE RESISTANCE OF TRUCKS AT DIFFERENT TEMPERATURES OF THE GEARBOX LUBRICANT

THE DISCUSSION

Major M. L. Ireland:—I would like to express the appreciation of the engineering division of the National Research Council for the very cordial cooperation that we received from Yale University in this work. We doubled the greatest pulling load that they had ever had on this apparatus. The maximum force at the tire surface illustrated by Professor Lockwood was about 600 lb. When we put the 7½-ton Mack truck on it, the pull ran to 1100 lb. The Quartermaster Corps' standardized Class B truck made it 1000 lb. The work done by Professor Lockwood was an essential part of our research because we were endeavoring to tear apart the group of forces that enter into what is known as tractive resistance, and it appears from a very careful survey that we have made of the literature of Western Europe and America on the subject of tractive resistance that there has been no really successful effort to separate that group of forces into its component parts, although Riedler has separated laboratory results, as distinguished from road results, into component parts. It is only by some such method as Professor Lockwood's, particularly when we are dealing with the gasoline-driven vehicle, that we can get at these results. When these results are published, the first thing that engineers will say is, "Why did you not run the speed test out to a greater range?" The answer will be that we went to the limit and beyond the capacity of the equipment. A survey of the equipment installed elsewhere indicates that no one has equipment that will withstand these big heavy trucks under full load and at high speed. One of the developments that I think the next few years will bring is the redesigning and rebuilding of apparatus similar to Professor Lockwood's to handle these heavy loads. Professor Lockwood is leading the way in a creditable manner.

Many of the best researches conducted in the past have wholly overlooked the matter of the degree to which the results are affected by the change of temperature in the differential. In scanning two or three researches that are regarded as practically the last word, we found that the investigators have not taken this into account, and that therefore their results may be affected seriously.

An interesting result in regard to this temperature-effect is this: we made six runs about 4 miles out of New Haven with the Mack 7½-ton truck, one run after the other, a total elapsed time of 82 min., with absolutely no change of any kind in the conditions other than the change in the temperature of the differential and, between the third and fourth runs, we opened the governor so that the speed at the crest of the hill was increased from about 11 m.p.h. to about 17, and the maximum speed from 18 m.p.h. to 21. When we came to take a mean curve for the first three runs, and a mean curve for the second three, we found altogether different results. Fortunately, there was a thermometer in the differential case that showed a change in the temperature of about 30 deg. fahr. According to Professor Lockwood's laboratory work of the day before, this would indicate a change of 15 m.p.h., or 11 tons in tractive resistance. When that correction is made on the chart at 15 m.p.h. it accounts very satisfactorily for the difference between the two final tractive-resistance curves. There is perhaps as much as 9 lb. per ton difference at the extreme ends and about 12 lb. in the middle, so that the laboratory work has borne out in a very satisfactory way many of the results that we are obtaining by actual measurement on the road.

The work of measuring tractive resistance covers a wide range of tires running over a wide range of loads. We have made tests on five different kinds of road thus far.

HERBERT CHASE:—Professor Lockwood's excellent paper represents an enormous amount of work. I found much useful information in the data that he has accumulated and prepared an article entitled Rear Wheel Dynamometer Tests and Their Significance to the Engineer[4] I had an opportunity to pick out, from some 200 tests that Professor Lockwood made, about 20 tests which seem to be fairly representative of the passenger-car field in particular and include also three or four trucks. The data that I secured are tabulated in the article referred to and the curves drawn for comparative purposes are shown therein.

The figures on wind-resistance in the table are computed, as already explained, by determining the projected frontal area of the car and multiplying the number of square feet by 0.003 and the square of the speed in miles per hour. This, of course, is only an approximation since the wind-resistance is affected materially by the streamlines of the body, but it serves well enough for comparative purposes. According to the figures given regarding these 15 to 20 passenger cars, the frontal area varies from 16 to 32 sq. ft. in the passenger cars listed. These areas no doubt could be decreased in some cases by improved body-design. Some means of actually measuring wind-resistance is needed before the actual resistance of certain other types and sizes can be predicted with certainty. Wind-resistance becomes an important factor at speeds as low as 20 m.p.h.

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[4] See *Automotive Industries,* April 20, 1922, p. 859.

The method of obtaining figures for tractive effort on level roads used by Professor Lockwood is to add to the wind-resistance the resistance of the front wheels of the car. This gives the pull that must be exerted at the periphery of the driving-wheels to propel the vehicle on a hard, level road at the various speeds indicated.

In testing cars on the dynamometer, the brake is set to give loads equivalent to the tractive resistance on the level road at the respective speeds, and the fuel consumption is measured. The two factors that oppose the forward motion of a car on a level road are the wind-resistance and the rolling resistance of the front wheels, the rear wheels and the elements connected to them.

The power delivered by the rear tires is the power that remains after overcoming the losses from the engine back to the rear wheels including the power lost in the rear tires, and that is precisely equal to W, the wind-resistance, plus R, the front-wheel resistance; R can be measured accurately by the method Professor Lockwood has described. The value of W is computed from the formula mentioned and is only an approximation. I suggest a more accurate method of obtaining the factor W. It is a method which will take into account the streamline form of the body; in short, measure its actual resistance. Quoting from my article

> The brake horsepower that the engine is required to develop for level-road operation at average driving speeds is small compared to its maximum power at these speeds. The average at 20 m.p.h. for 17 passenger cars is about 5.4 hp. as against a maximum average at the same speed of 23.8 hp. In other words, under what can be considered a normal operating condition, the average engine develops from one-fifth to one-quarter of the power it is capable of developing at that speed. Even the heaviest of the passenger cars tested requires only 7.55 b.hp. to propel it at 20 m.p.h. on a level road, while some of the lighter cars require only a little over 4 b.hp. It seems almost incredible that if facts such as

> these had been more generally appreciated, so little attention would have been given to making engines more efficient at the light loads obtained in normal use.
>
> It is instructive to note from the tabular data the relative importance of the three losses, which together exactly balance the power developed at the flywheel of the engine. These are (*a*) the rolling resistance of the front wheel, (*b*) the power losses between the engine and the driving surface of the rear tires, including the losses in these tires, and (*c*) the wind-resistance.

The method of measuring the brake horsepower, in the test that Professor Lockwood makes, is to drive the rear wheels from the drums to which power is applied by the electric motor below the floor. The power is transmitted through the tires and the bevel gears to the upper shaft of the transmission. The gears are placed in neutral. There is some loss due to the churning of the lubricant in the gearcase. There is also a loss in the gearing, which may or may not be the same as that which occurs when the engine is driving the car under load. In other words, the gears are running light as compared to running under load when driving the car. Furthermore, there probably is a certain difference between the power absorbed by tires that are merely rolling and tires that are driving. For these reasons the power losses thus secured, when added to the rear-wheel horsepower, probably do not give exactly the brake horsepower of the engine, but for comparative purposes they are doubtless good enough. If Professor Lockwood had the equipment, it would be interesting to know just how much greater this loss would become if the gears and parts that transmit the power were actually loaded.

In regard to wind-resistance, a suggestion is made in reference to a method of measurement, more accurate than using the computation based on the frontal area. To do this the car should be equipped with a vacuum gage, connected to the inlet-manifold, and an air-speed meter. It would then be taken to a stretch of level road and operated at the desired speeds, checked by speedometer and stopwatch. In each case the reading of the air-speed meter and the vacuum gage would be noted, the former indicating any head or tail wind that might affect the results. The car would then be brought back to the laboratory and tested at the same speeds and throttle positions, as indicated by manifold depression, and the rear-wheel power measured. The difference between this rear-wheel power and the power absorbed by the front wheels would be the actual power used in overcoming wind-resistance at the respective speeds.

To secure a more certain determination of the brake horsepower, and consequently a more accurate method of measuring the losses between the engine and the rear wheels, requires equipment that may not now be available in the Mason Laboratory and that may not be as easily connected up and used as the present equipment. I am under the impression that it would be possible in some cases to make a connection to the front end of the engine in the place of the starting-crank, and by a light prony brake, or some other form of brake, measure the brake horsepower of the engine directly. Measuring the difference between the brake horsepower and the rear-wheel horsepower, as it is now measured, would afford a means for precise determination of the losses between the engine and the periphery of the rear tires.

The losses in the rear tires can probably be assumed to be approximately equal to the losses in the front tires, if the tires themselves, the inflation-pressures and the weights on the tires are the same, as they would be for purposes of calibration.

E. Favary:—Will Professor Lockwood give particulars regarding the difficulties he experienced with the drawbar-pull tests of his earlier dynamometer?

As to the losses in tires, some of the results I found a few years ago probably hold good at present. Professor Lockwood showed that a partly worn solid tire had less rolling resistance than a new solid-rubber tire. This is easy of explanation when we consider that the tests are made on a smooth drum. If the road surface were perfectly smooth, a solid-steel tire would, I think, show the least resistance. However, the rougher the road is, the softer must be the pneumatic tire to show the least rolling-resistance, because the most perfect tire is that which will absorb the average height of obstructions on the road without raising the axles. The solid tire has an increased loss on account of the low elastic efficiency of rubber.

Prof. E. H. Lockwood:—Mr. Favary has raised an interesting question on the relative resistance of solid-rubber and pneumatic tires on smooth and on rough surfaces. His doubts that solid-rubber tires on rough roads would roll as easily as fabric pneumatic tires seem well justified. Experiments are now in progress in three different States for road resistance of loaded trucks on smooth portland-cement roads. Preliminary figures from all of the tests appear to indicate that solid-rubber tires have actually lower rolling resistance than fabric tires on passenger cars. Data are not at hand for badly worn roads, where doubtless the solid-rubber tires would offer relatively more resistance.

THOMAS S. KEMBLE:—Has Professor Lockwood been able to determine the average overall efficiency of passenger-cars and trucks?

PROFESSOR LOCKWOOD:—Questions about efficiency are difficult to answer for the simple reason that the efficiency of a vehicle has not been defined. I should therefore begin by defining the term as the ratio of the power delivered at the periphery of the rear tires to the power developed at the clutch. According to this definition, if the friction losses between the clutch and the rear tires could be reduced to zero, the efficiency would be 100 per cent. On the other hand, if the engine is idling with gears in neutral, the efficiency will be zero, because in that case no power is delivered to the rear tires. Average efficiencies under normal operating conditions can be computed for typical vehicles, as shown in the several examples chosen from tests at the Mason Laboratory that are given in Table 15.

It appears that the efficiency as defined may extend from zero to a maximum of 90 per cent. It is gratifying to see that an efficiency of 90 per cent can be realized, but it must be remembered that under ordinary running conditions the figure is much lower, say 50 to 60 per cent.

MR. KEMBLE:—Did you give the tractive resistance of the front wheels separately from the rear wheels? Is your comparison of the cord, fabric and solid tires for the front or for the rear wheels? It seemed to me that it was for the rear wheels and, unless there is some further explanation, we might be led astray. There are transmission losses which, if included, affect the percentages; whereas, if the comparison were made on the front wheels, the loss in the bearings being comparatively small, the percentages would be more accurate.

PROFESSOR LOCKWOOD:—That is an important point. When we measured the frictional resistance of pneumatic tires, whether on the front or rear wheels, we invariably got the same results. Ordinarily, the front-wheel bearing-resistance is from 4 to 6 lb.; the rear, including transmissions, from 15 to 20 lb. This resistance, sub-

TABLE 15—AVERAGE MOTOR-VEHICLE EFFICIENCIES UNDER NORMAL OPERATING CONDITIONS

Vehicle	Kind of Tire	Tire Size, In.	Weight, Empty, Lb.	Speed, M.P.H.	Efficiency, Per Cent	
					Level Road	Steepest Grade
Light Touring Car—Dodge.......	Cord	32x4	2,655	20	41 ÷ (41 + 25) = 62	295 ÷ (295 + 30) = 91
Heavy Touring Car—Cadillac.....	Cord	35x5	4,395	20	61 ÷ (61 + 56) = 52	577 ÷ (577 + 53) = 90
Heavy Truck—Standard, Class B..	Solid Rubber	....	11,540	15	111 ÷ (111 + 127) = 47	588 ÷ (588 + 135) = 81

tracted from the total, leaves the difference we call tire resistance. We believe that the tire resistance is the same whether measured on the front or the rear wheels. Our tests have been made on the rear wheels, as a matter of convenience in changing the loads.

MR. KEMBLE:—You take as the tire resistance of the rear end the difference between the resistance with the load on and the resistance when the wheels have been raised so that they just touch the drum; has that been checked to a certain extent with the resistance that you get on the front wheels?

PROFESSOR LOCKWOOD:—Yes.

C. F. SCOTT:—Is the large-diameter drum essential to successful results? Could satisfactory results be expected with, say, 3½-ft. drums instead of 5-ft.; and with steel drums?

PROFESSOR LOCKWOOD:—I do not feel very positive on this point. Whether it could be predicted that the small-size drum would give different results, I am not prepared to say. It would be an interesting experiment to have the same vehicle tested on the steel drums at Worcester, and then on the Mason Laboratory drums, to see whether the results in one place would be the same as those in the other. When Mr. Favary's apparatus at Cooper Union, New York City, is completed, it would be instructive to have it compared with the others.

MR. SCOTT:—To repeat Mr. Favary's question, why did not the drawbar measurements work out?

PROFESSOR LOCKWOOD:—Like most experimenters, we tried to measure this resistance directly. The first thing we discovered was that the front wheels, resting on the floor, varied the drawbar pull by from 15 to 20 lb. To avoid this error it would be necessary to raise the front tires, allowing the front end to roll on anti-friction rollers, as was done by Riedler in Germany.

We tried to suspend the front end with our overhead crane but found difficulty with side-sway and abandoned the trial. Another thing is the difficulty of keeping the contact of the rear wheels directly over the center of the drums. If it gets slightly off-center, there is a gravity component which has some effect. Due to such troubles, we discarded the attempt to measure the drawbar pull directly and used the friction-brake pull instead.

R. E. PLIMPTON:—How much would the frictional resistance increase with the diameter of the rear wheel, if the load were constant; say with tires between 34 and 40 in.?

PROFESSOR LOCKWOOD:—It is my offhand impression that it would not make much difference. I do not know of any experiments to determine the effect of tire diameter on the rolling resistance on smooth roads. I suspect that the effect produced by any change of the diameter is very slight, perhaps too small to measure.

CORNELIUS T. MYERS:—The old subject of the gasoline engine working at low efficiency in passenger cars and trucks is still with us and probably will be for some time. In 1910 and 1911, in the design of a new line of trucks, it was very thoroughly threshed out in one instance, and an effort was made to impose on the sales department a line of trucks with much smaller engines than had been considered good practice up to that time. We encountered considerable "rolling resistance" in the sales department, but a tractive-factor formula was developed[5] which took into consideration the various factors that Professor Lockwood has set forth.

In this tractive-factor formula the resistance was measured in fractions of a pound per pound of weight. Professor Lockwood has suggested that instead of using pounds per ton, we use pounds per thousand pounds. I should like to go a little farther and use a plain pound coefficient. It will be a decimal of a pound, of course, per pound of weight. That is directly comparable with the grade coefficient as say 2 per cent, or 0.02 lb. per lb.

PROFESSOR LOCKWOOD:—It seems to me 100 would be more logical. Then 10 lb. per M would be 1 lb. per 100 lb. Why not take 100?

MR. MYERS:—For the reason just stated. In going into the subject I found nothing whatever in this Country, but in France and Italy the road coefficient used is kilograms of resistance per kilogram of weight. It is understandable and is the ordinary method of expressing a coefficient.

The wind-resistance for any car or truck can be determined indirectly. With how great accuracy I cannot say because I have not made enough experiments. In looking at Professor Lockwood's illustration, we see that if the car were allowed to coast, after it reached a certain speed there would be the resistance of the front and rear wheels and of the axle and transmission, if the gear is in neutral and the engine shut-off.

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[5] See S. A. E. HANDBOOK, vol. 2, p. 67.

Professor Lockwood has obtained what we have had heretofore only approximately from Riedler's experiments. He has given us the resistance of the tires. One thing is left and that is the wind-resistance. You can get that from the accelerometer readings, from which the above resistances must be deducted. When the car is run in opposite directions, with and against the wind, if the gear is placed in neutral and the other four resistances are deducted, the wind-resistance can be determined. It is a definite and understandable figure. In 1912 and 1913 we checked the rolling resistance of cars with the accelerometer and found, as Professor Lockwood did, that if the truck is in good condition, the rolling resistance is about 40 lb. per ton and that of the passenger car is about 30 lb. per ton at 15 m.p.h.

M. C. HORINE:—We have talked about the mechanical efficiency of the vehicle, the wind-resistance and the tire-resistance. There is another factor, the road-resistance. Professor Lockwood's tests were performed under ideal conditions so far as the road is concerned. Still another is the friction generated by the spring-suspension of the vehicle. It takes power to lift a vehicle bodily into the air and let it down again. That is the acceleration and deceleration of the mass. There is friction in the springs and the spring-shackles. It takes power to move those parts, although the power is not applied directly. The motion of the parts and the wear that results cause the consumption of driving power. Consequently, more than a laboratory test is needed to determine the exact amount of power required to move a car over a given road. Has Professor Lockwood considered the matter of springs in the power losses in the suspension of the vehicle?

PROFESSOR LOCKWOOD:—That is a topic which is rather new to me. I am not sure that it would not be deserving of attention. The measurement of it would be a nice problem.

MAJOR IRELAND:—We are getting some interesting results from road tests, but it is early yet to give definite predictions, much less laws and formulas. One interesting test that we happened to make was on some granite blocks between street-car tracks down a certain hill. We did not know at the time that we had crossed the line between the cities of Everett and Malden, Mass. In Everett the blocks have cement grouting between the joints and the road surface is smooth; on the other side of the city line in Malden, the joints are sand-filled. There is a marked difference in the way some of these granite-block sets stand above one another in the two cities, and there is a marked difference in the resistances at the same velocity in two parts of the run. We have photographed several sections of the road and measured the degree of roughness with a 10-ft. straight edge.

If the amount of energy absorbed in the spring-suspension, frame, tires and other vehicle parts is subtracted, there would be included in the remainder that part of the road-test results which is due to the road itself, and this would show in a much clearer way the comparative effects of the sand-filled and grout-filled joints. The separation of the tire-resistance and the resistance of the transmission system is not difficult. How to separate the effect of the increased impact absorbed by the vehicle parts from that which is absorbed in the road material and therefore belongs in what is called road resistance is a problem. About the only means of doing this simply seems to be to deduct very carefully all tire and transmission-system resistance that is due to other causes, and then, from the principle that action and reaction are equal but opposite in direction, divide the remainder, which should be impact effect, into one-half absorbed by the vehicle parts and one-half absorbed by the road material. It will be seen that, in practice, some serious difficulties will be encountered in doing this.

AUSTIN M. WOLF:—The results obtained in these tests would be comparable to an ideal roadway. Besides the energy expended in the inter-leaf friction and the friction in the spring-shackles, much energy is absorbed in the moving of the great mass represented by the weight on the springs. The universal-joints are working at greater and irregular angularities; the tires are flexed to a greater extent than on the dynamometer drum; the air within the tires is compressed whenever a bump is met; and, in addition, there is the side-slipping and the waste of energy caused by it. After striking an obstacle, one wheel that is in the air is speeded up through the differential and when the tire returns to the ground the tread is chafed. All these factors may be small severally, but collectively they represent a large amount of energy.

Taking these things into consideration, Mr. Chase's method of measuring air-resistance would give a result too high, as the losses indicated on the dynamometer would be less than those actually encountered on the road.

I suggest that in future tests an attempt be made to use a dynamometer drum having a detachable tread so that various irregularities in the road could be represented. This would be somewhat similar to the method used some time ago in attempting to run a destruction test on aluminum wheels. It would be interesting to have each rear wheel rest on an independent drum and the speeds varied to reproduce the action of a car in turning a curve. The losses due to the differential might thus be determined.

It probably would be difficult to make accurate measurements on a dynamometer with a detachable track on the drums, but I believe that a careful analysis of the problem would result in adequate equipment. To get more accurate data in road tests, it occurred to me long ago that we are not measuring the development and expenditure of energy by a simple and direct means. I suggest that the engine be cradled in the chassis and the torque reaction be measured directly in this way; similarly, with a separately mounted transmission it also can be cradled and the torque reactions noted, when in other than high gear. With a torque-arm attached to the rear axle, the reaction at its point of anchorage on the chassis can be measured. All these parts should work on a recording type of instrument, as it would not be practicable to take visual reading at all these points during a test.

Professor Lockwood:—The point raised by Mr. Wolf that the losses indicated on the dynamometer would be less than those actually encountered on the road, seems reasonable. It is surprising, however, that for the few vehicles thus far tested on the Mason Laboratory dynamometer in comparison with new cement roads, the road-resistance appears to be somewhat lower. It must be remembered that this comparison involves the deduction of the wind-resistance from the road results, and thus far wind-resistance has been computed by a formula and is rather approximate.

The dynamometer results are less ideal than might be expected. The Mason Laboratory drums are not perfectly round. The variation of the radius at present is about 1/16 in. Due to this and variations of the tires themselves, most cars run on the drums with a slight up-and-down motion, which undoubtedly absorbs power just as on the road.

NEW AUTOMOTIVE-VEHICLE SPRING-SUSPENSION[1]

By H M Crane[2]

The author indicates what the history of spring-suspension has been but discusses only the conventional type of four-wheeled design in which the front wheels are used for steering and the rear wheels for driving and braking. The problem of front-axle spring-suspension is mentioned, but that of proper rear-axle spring-suspension, especially for passenger cars, is discussed in detail because it is a much more difficult one.

The advantages of the Hotchkiss drive for shaft-driven cars and some of its distinct disadvantages are stated, shaft-driven, rear-axle mountings being commented upon in explaining the factors that influenced the design of the spring-suspension device developed by the author. The advantageous features of this device are enumerated, inclusive of the effects of tire reactions.

The first wheeled vehicles of which we have any record were not provided with any special spring members for the cushioning of road shocks. They were not entirely springless, however, for a certain degree of cushioning effect was furnished by the natural resilience of the materials used in the construction of the various parts. There are many horse-drawn wagons still in use today which embody this type of design. As road transportation became more highly developed and speeds increased, vehicles were produced in which the spring action of the various parts was increased by modification of the design, but still without the use of special spring members. The buckboard and the stage coach are instances of this stage of development.

Near the end of the last century, however, the art of spring-suspension, using steel leaf-springs of various forms, had reached a considerable degree of development, at least for horse-drawn vehicles. The early designer of automotive vehicles therefore had a valuable fund of information on this subject with which to work. This information could be used only in conjunction with the working out of an entirely new set of problems, imposed by the fact that the automotive vehicle is propelled, stopped and steered by means of its wheels. For the purposes of this paper, it will be necessary only to discuss the simple and conventional type of four-wheeled design, in which the front wheels are used for steering and the rear wheels are used for driving and braking.

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In front-axle design, it is almost universal practice to pivot the wheels for steering by mounting the short stub-axles or knuckles, on which they revolve, on the ends of the axle by substantially vertical hinge-pins. The axle itself requires only to be connected to the frame in such a manner as to remain in a plane approximately at right angles to the longitudinal center-line of the chassis and so that a linkage can be arranged between the steering-gear, which is fixed on the chassis frame, and the steering connection on the axle, in a way that will allow of the axle moving relatively to the frame without tending to alter the direction of the front wheels. These requirements can be met fairly well by using a pair of semi-elliptic springs, fixed at one end and shackled at the other and, due to its simplicity, this is the favorite arrangement today. As I will explain later, this system is not ideal from the point of view of absorbing road shocks by the springs. It depends to a considerable extent for its success upon the pneumatic or other cushion tire, as well as on the fact that, in passenger cars, where easy riding is important, the passengers rarely are carried on the chassis at a point forward of the center of the wheelbase.

REAR-AXLE SPRING-SUSPENSION

The problem of a proper rear-axle spring-suspension, especially for passenger cars, is a much more difficult one. Like the front axle, the rear axle must be held in its correct position with respect to the frame, but it also must be supported in such a way that the rear wheels can drive or stop the car as required. Furthermore, the connection between the axle and the frame should be designed to minimize the transmission of road shocks from the former to the latter. A brief outline of the history of automotive design in this regard will help to make clear the questions involved.

In most of the early cars, the power was transmitted to the rear wheels by chains, either a single central chain or double side-chains being used. Let us consider only the latter type of construction, which clearly illustrates the principles of design. Rigid distance-rods are used between the axle and the frame. These are required to preserve the correct center distance between the driving and driven sprockets, and also serve to maintain the axle in a proper position with relation to the frame. Semi-elliptic springs, shackled at both ends, support the body load and also act to hold the axle laterally. In driving and braking the vehicle by the wheels, torque reactions are set up. In the type of design under discussion, the reactions due to driving or to the use of a transmission brake are taken up in the jack-shaft housing. The reactions from the rear-wheel brakes are taken up in the springs sometimes, but more often by the distance-rods, which are then made stiff enough to act as torque-arms. In the chain-driven car, the rear axle is relatively light and has therefore no great tendency to produce shocks in the vehicle structure. This is not the case with the much heavier axle of the shaft-driven car, and this fact must be recognized in the design of the latter type if the best results are to be obtained.

The early shaft-driven cars tended to follow chain-driven designs, using distance-rods, and adding a torque-arm attached to the rear-axle housing and supported at some point on the frame to take care of the torque reactions arising from driving and braking. The springs were used only as supporting members, being usually mounted so as to turn freely on the axle, and being shackled at both ends. In striving for greater simplicity and decreased weight, it was found that the distance-rods could be dispensed with and their functions performed by the front halves of the rear springs, the shackles at the front ends being abandoned and plain pivot-pins used in their place. A further move in the direction of simplicity led to the Hotchkiss drive, in which system the rear axle is entirely controlled by the rear springs. The springs support the load, maintain the axle in correct relation with the frame and absorb all torque reactions, whether due to driving or braking.

The Hotchkiss drive for shaft-driven cars has some very manifest advantages. It is light in weight and has a minimum number of parts to wear and rattle. It gives a cushioned resistance to suddenly applied driving and braking loads. Moreover, if the springs are properly proportioned, it maintains the center-line of the pinion shaft substantially parallel throughout the range of axle movement. This latter feature is of an importance not often recognized. If a rear-axle torque-arm is used, whenever the axle approaches or recedes from the frame the axle housing must rotate through an angle depending upon the amount of movement and the length of the torque-arm, the angle being greater the shorter the torque-arm is. If there is no lost motion in the parts, the pinion shaft must rotate to an amount depending on the angle through which the rear-axle housing turns, and on the gear-ratio. This would not be a serious matter if the pinion shaft alone were involved but, when the car is being driven by the engine, the pinion shaft is connected through the driving-shafts to the engine flywheel, which is a part of the system designed to turn as nearly as possible at a constant angular velocity. This combination of circumstances is capable of producing shocks in the driving members which are far greater than will be believed readily by those who have not observed this action.

The Hotchkiss drive has also some distinct disadvantages. When the rear springs are sufficiently flexible to give good riding, the resistance to torque reaction is too soft, with the result that, especially in lighter cars, the rear axle tends to jump when under heavy driving or braking loads and on soft or rough roads. From the point of view of good riding it leaves much to be desired. It furnishes a sufficient degree of flexibility in a substantially vertical direction, but is far too rigid otherwise.

Let us consider any of the present common types of shaft-driven rear-axle mounting, either of the Hotchkiss design or of designs using torque-arms or distance-rods, or both in combination. In practically all of these arrangements the axle is positioned rigidly with respect to the frame in a longitudinal direction. If the weight of the axle were relatively small, this rigidity would not be a matter of serious moment. The fact is, however, that the weight of the rear axle and wheels of most shaft-driven cars approaches 10 per cent of the total loaded weight of the vehicle. The ideal of riding is to have the spring-borne parts of the vehicle, the frame, the power-plant, the body and the load, follow the major contour of the road surface without being affected by minor undulations. It would be desirable to have this condition apply also to the parts below the springs, the wheels and axles, and the pneumatic tires help to attain this result, but there is a limit to their ability in this direction unless made of extreme dimensions. Practically, the axles and wheels follow a very irregular course, with rapid and

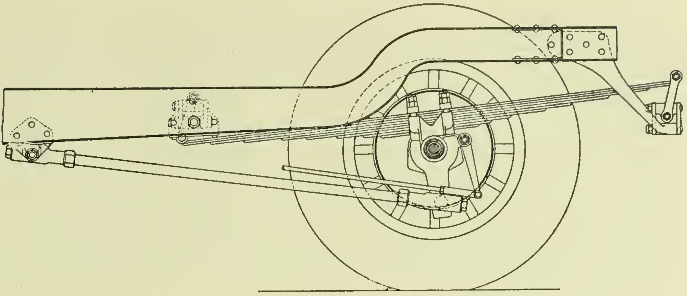

FIG. 1—SCHEMATIC DRAWING OF A RECENTLY DEVELOPED SPRING-SUSPENSION

frequent changes of direction in a vertical plane. To change the direction of motion of a moving mass force must be applied and, if the mass is great and the change in direction sudden, that force must be large. In most cases the forces in question are the reaction of the road surface through the tire and the horizontal attachment between the axle and the frame such as distance-rods or springs. This latter connection is usually extremely rigid, and so it follows that the only cushioning for the shocks occasioned by these forces is furnished by the tires and what little resilience there may be in the road surface. Spring-cushioned distance-rods have been tried with a view to ameliorating this condition, but the cure in this form has seemed to be worse than the disease. Full-elliptic springs used in connection with the Hotchkiss drive are not greatly open to the above objection, but have the defect of being much more badly adapted to absorb torque-reaction loads than are semi-elliptic springs.

THE CRANE SPRING-SUSPENSION

With the idea of combining the simplicity and other advantages of the Hotchkiss drive with the proper cushioning of the shocks just described, the device illustrated in Figs. 1 and 2 was conceived. In addition, it appears in practice to remove two of the principal disadvantages of the Hotchkiss drive, by greatly increasing the resistance to torque reaction of any given set of springs without entirely removing the desirable cushioning effect to driving and braking shocks and, because of this fact, doing away almost completely with the jumping action of the rear axle on soft roads, already criticized.

Fig. 1, which is a schematic drawing of the arrangement, shows a semi-elliptic spring, shackled at both ends, and rigidly bolted to the axle. It shows also a distance-rod, the forward end of which is connected to the frame by a ball-and-socket joint and the rear end is connected to the axle by a similar joint, not at the center-line of the axle, but at a point substantially below the center-line, a bracket rigidly attached to the axle being provided for the purpose. For clearness, only one end of the axle is shown on the drawing, the equipment of the other end being the same.

It is obvious that, with this arrangement, the axle is free to move longitudinally with respect to the frame as well as vertically, while at the same time it is constrained in both these movements by the resistance of the semi-elliptic spring. The vertical resistance need not be explained, being the normal spring action. The resistance to longitudinal movement is occasioned by the fact that, to translate longitudinally, the axle casing must rotate about the ball-and-socket joint located below the center of the axle. This rotation is resisted by a couple set up in the spring, tending to increase the load on one arm of the spring and decrease it on the other. The character of this resistance evidently can be varied by changing the distance of the point of attachment of the distance rod below the center of the axle. The spring shackles are made long enough not to interfere with this action. It is also necessary to provide for extra longitudinal come-and-go in the propeller-shaft connection.

In the conventional Hotchkiss drive the torque reactions due to driving and braking are resisted by couples set up in the springs. This is still the case in this new arrangement, but the couples are of considerably less magnitude due to the action of the lever-arms to which the distance-rods are attached. The reduced twisting of the axle is very evident in actual driving, as is also the improved action over soft rough roads when using heavy power. The latter improvement undoubtedly is caused by the better method of taking care of the torque reactions and by a different type of tire reaction with this suspension.

TIRE REACTION

This question of tire reaction is a very important one when the axle weight is high compared to the total weight of the vehicle. The pneumatic tire, especially one of the cord type, constitutes a most efficient spring. In other words it gives back practically everything that is put into it, with a minimum of damping action. The tossing about of such a weight between the tire spring on one side and the vehicle spring on the other certainly interferes with the smooth riding of the vehicle as a whole. This is why it is desirable to reduce the axle weight to the lowest possible figure. Unfortunately, it is not feasible in the case of the shaft-driven axle, even with the most careful design, to reduce the weight sufficiently to allow of ignoring it altogether. The new method of suspension, just described, is believed to provide a considerable improvement over conventional arrangements by reducing materially the maximum compression in the tire under any given set of operating conditions, and by giving an attachment between the axle and the frame that is cushioned in all directions except laterally.

One cause of excessive tire compression was partly explained in a previous paragraph that outlined the forces acting on an axle rapidly traversing a rough road. This action can be understood most easily by examining the action of an axle crossing a hump on the roadway. If rigid distance-rods are used, the momentum of the axle in a horizontal direction is augmented by that due to the total car-weight to a degree depending on the angle and the height of the hump. The reason for this is that the line of force through the wheel to the wheel hub and so to the axle is inclined at a considerable angle by the action of the hump, although this line is substantially vertical when the wheel is traversing a smooth level surface. The inclined force resolves into vertical and horizontal components at the axle center. The opposing forces are, vertically, the weight of the axle and wheels plus the pressure of the springs and, horizontally, the total weight of the vehicle. At low speeds, the momentum effect of the masses is relatively small. At high speeds, it is very great.

In a previous paragraph I called attention to the fact that the flat semi-elliptic spring is deficient in flexibility except in a vertical direction. This is due to its stiffness as a beam in a lateral direction and to its great torsional stiffness. The use of rebound clips, tying the plates more or less rigidly together, is a contributing factor to the foregoing. If all irregularities of the road surface were symmetrical and were crossed at substantially right angles by the axle and the wheels, the lack of flexibility in directions other than the vertical would be of no importance. Practically, however, the contour of the road surface traversed by the wheels on one side of a car is rarely the same as the contour of the surface traversed by the wheels on the opposite side. This fact in itself tends to cause lateral shocks. These are augmented by the constantly changing relation between the axle and the floor of the car which do not remain parallel but vary from parallelism by considerable angles in either direction. Such changing angular relation can be accomplished only by the springs or frame, or both, flexing both torsionally and laterally. Of course, this action tends to reduce what is commonly called "rolling" on curves and heavily crowned roads, but it does so at the expense of some severe strains and shocks in the parts involved, which produce an unpleasant jarring effect in the riding on cobble stones and similar road surfaces. It is the action just described that makes it so difficult to keep the rattles out of some of our modern passenger-cars. Actually, a very few thousandths of an inch of side-play of a spring in a shackle is sufficient to cause a most unpleasant noise.

Fig. 2 shows one method of improving this action when semi-elliptic springs are used on a passenger-car chassis. The springs are connected to the chassis frame at both ends by short links that allow of practically unlimited twisting and a limited amount of side swing. It is possible that this arrangement may give rise to an excessive amount of rolling, but tests so far made, with the chassis loaded as shown in the photograph, do not indicate this. To minimize the chances of rolling, the springs are carried as high as possible, being mounted above the axle. The springs are tilted down in front, so as to clear the doors and floor of the body. It is probable also that stiffer springs, that is, springs having a steeper scale in pounds per inch of deflection, can be used because of the better all-round cushioning effect obtained. One advantage of the full-elliptic type of spring is its better action in cushioning lateral shocks. If it were not for its weakness in resisting axle torsion, a lack of faith on the part of designers in regard to its ability to position properly the steering axle in heavy-duty service and its interference with low-hung bodies when used for rear-axle suspension, it might well become a popular type for passenger-car use.

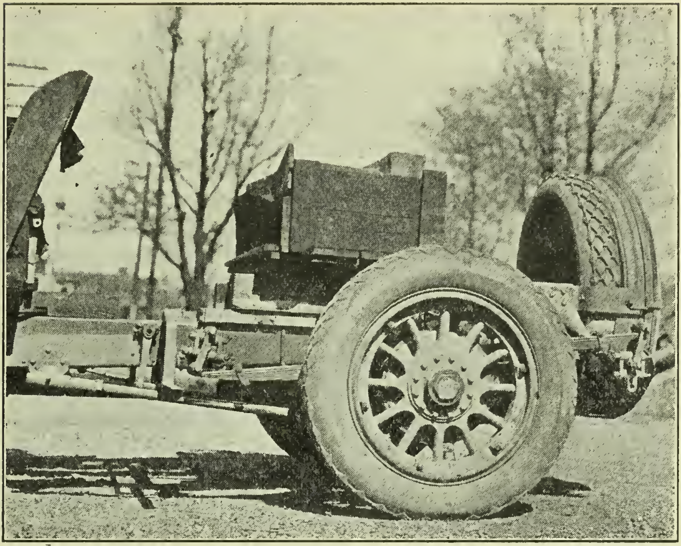

FIG. 2—APPLICATION OF THE SPRING-SUSPENSION ILLUSTRATED IN FIG. 1 TO A PASSENGER-CAR CHASSIS

The cord pneumatic tire, due to the low air-pressures that can be used successfully with it, has been the greatest single factor in improving the riding of automotive vehicles in recent years. The object of this paper is to call attention to the fact that there is still much that can be improved in the design of the other parts contributing to riding comfort.

THE DISCUSSION

H. M. CRANE:—The history back of this spring-suspension begins with a shaft-driven car equipped with distance-rods, a torque-arm and springs that were free to turn on the rear axle. This construction was followed by a practically similar construction in a somewhat larger car, about 1907. In the second construction we first had our attention called to the effect of the torque-arm in causing a reaction on the driveshaft, as described in the paper. An experiment we made there to ameliorate this condition consisted in providing a very considerable spring-resisted motion at the end of the torque-arm; that is, the fixed end on the chassis frame. Examination of this spring action showed that, over a comparatively smooth road, with the engine just pulling the car at about 25 m.p.h. on the level, you would occasionally bottom over 200 lb. of spring pressure on a torque-arm about 45 in. long. It gives a crude idea of the shocks that are produced in the driving mechanism by this effort of the rear-axle pinion-shaft to remain parallel to the gearbox shaft at the forward end. It is evident that any angularity that the rear-axle pinion is forced to take in relation to the fixed shafts on the chassis frame tends to result in an angular rotation of that shaft, or of some part of the driving mechanism; and the shaft is about the only thing to take it up.

I should have called attention in the paper to the fact that the use of fabric universal-joints, which is becoming very common at present, will go a long way toward eliminating the difficulties arising from this type of reaction; but the answer that we made to it at the time when it came up was to adopt the modified Hotchkiss drive. In other words, we clamped the rear axle solidly to the rear springs, proportioned the rear-spring arms so that we had a practically parallel motion between the axle and the frame, and used in addition distance-rods to align the axle and prevent it from going out of position. We also used a platform spring, overhung on the axle. That spring is equipped with a very satisfactory device for cushioning side-shocks in its cross-shackles, as I now know from having departed from it. As many of us have often done in attempting to improve, we actually arrived at an arrangement that was less satisfactory than the one we had been using previously.

Our next movement from this device was to the semi-elliptic spring hung under the frame, about 1912. The first cars of this type had very light frames and were driven almost entirely as open cars. The thing in this particular design which began to cause us to open our eyes was a breakage of the rear-frame member. The frame was designed at the rear end very much as at the front end, and we broke off the overhanging part of the frame beyond the last cross-member. That was due to the lateral vibration set up by these very flat springs that were stiff laterally, and partly to the twisting of the springs tending to twist this rear extension of the frame. However, we believed that by making the frame sufficiently strong there we would overcome this difficulty, and we proceeded to a larger car of 142½-in. wheelbase, using the same general layout of a flat semi-elliptic underhung spring at the rear. The riding quality in the rear seat of this car was very disappointing at the start. The difference was not particularly noticeable in the front seat, but there was a disagreeable lateral chattering effect that could be felt on the floor-boards and the cushions and showed up very plainly on the battery hanging. The battery was suspended under the rear floor in a hanging of fairly light weight and not at all rigid, and the battery was trembling sidewise sufficiently to shake the whole floor. It was at that time that we introduced the spring cushioning on each side of the rear spring at the front end and the ball shackle at the rear end; that is, a shackle in which the spring end and the shackle connection were of the usual form, but at the other end of the shackle was a ball-joint attachment to the frame. This allowed a considerable amount of lateral freedom to the spring and at the same time reduced the torsional reaction of the spring against the frame when one wheel was lifted higher than the other. It was a fair compromise but not entirely satisfactory.

This car had a straight Hotchkiss-drive; in other words, there were no distance-rods. All of the connection between the rear axle and the frame was made through the spring; the driving torque-reaction, brake reaction and everything. We have found that the springs are entirely capable of doing this satisfactorily; that is, so far as spring breakage is concerned. To give figures, the usual flexibility of rear springs on this car was of the order of 165 to 170 lb. per in. of deflection for each spring. This is on a car that weighed, without passengers but otherwise fully loaded, from 5000 to 6000 lb. The spring, relatively, was of a very soft character.

In driving many miles with chassis of this type, and even driving with cars with bodies on, with the floor-boards up and watching the rear-axle operation, it became evident that there were other conditions there that were not being met. In other words, we had a very heavy axle tied to the frame in a vertical direction, with a considerable degree of flexibility; but, longitudinally to the frame, the springs being practically horizontal, there was almost no flexibility whatever. We ought to have realized this from the fact that we broke the front spring-eyes of a great many springs, in this case when we were using 1¼-in. bolts at the front end of the spring, the spring eye not being welded, but simply turned over and acting as a hook. We found it necessary to reduce the diameter of the bolt to 1 in. to make the spring stand up.

I began to feel that there ought to be some way of letting the rear axle accommodate itself to a certain extent in speed to the longitudinal-horizontal speed of the chassis. That had been done before and there was no question about it; the thing had been recognized as a desirable feature and, as I stated in my paper, there have been springs introduced in distance-rods, or the springs have been canted at an angle with the horizontal so that the rear axle would tend to move toward the rear of the car when the springs were compressed, and there have been a number of similar ways in which there has been an effort to overcome this defect that was generally recognized.

Some of the defects of the Hotchkiss drive have also been recognized. It is easier on tires, I am very sure, on ordinary roads, but we all know the effect that is produced on sandy, gravelly hills with the lightly loaded Hotchkiss-drive job; there is a tendency for the rear axle to jump clear of the ground and set up a vibration period in jumping and in rotation that necessitates an entire abandonment of any attempt to go up the hill rapidly, and simply necessitates reducing the application of power until it is just barely possible to crawl up the hill.

In thinking these various things over, I evolved the layout that is shown in Fig. 1. The first thought I had was: Suppose we use a distance-rod but attach it at the level of the ground; that is, opposite the tread of the tire on the ground. Apparently that would practically eliminate the torque-reaction effect; in other words, we would transmit it directly to the frame through the distance-rod. At the same time I saw that if we attached the distance-rod at a point not at the center of the axle but at a point substantially below the center, it would be possible for the axle to travel longitudinally with respect to the frame to an extent permitted by the stiffness of the spring itself. The arrangement was first tried on a car of about 4200-lb. weight and, as an experiment, the point of attachment was placed at about half the distance from the center of the axle to the ground. In this job we were able to vary the position of attachment vertically so as to get a longer or a shorter lever-arm; and our experiments covered lever-arms rather more than half of the radius of the tire. The results of the device were two. First, there was a very noticeable and continuous variation of the position of the axle with relation to the frame in a horizontal direction on any kind of road, even a comparatively smooth, newly built highway. On a rapid reversal in a vertical direction, such as a "thank-you-ma'am" or a gutter in the pavement, there was a very violent and extended movement of the rear axle in the longitudinal direction and at the same time a very much reduced vertical throw of the body. I have tried to explain in my paper why I think that reduced vertical throw occurred; that it is due to the reduced tire-reaction. It is very difficult to tell exactly what is going on under these conditions, but I think that is the reason. As to the facts, there was no doubt whatever; there was a very great difference.

Another thing we found in an early application was that if the longitudinal stiffness was made too small by lengthening the lever-arm, it was wholly possible to obtain a serious periodic vibration longitudinally of the axle with respect to the frame on a washboard road; in other words, the axle would move backward and forward violently in a period determined by the stiffness of the spring reaction and the weight of the axle itself.

I am not convinced yet that we know exactly the best length of lever-arm to use with the arrangement. It must depend very largely on the weight of the axle compared to the stiffness of the spring and to the weight of the vehicle itself, but it is apparently possible to obtain a relation of parts that will practically never cause this unfortunate effect, and it certainly is most unpleasant when it occurs.

What I have just described is the new part of this suspension. I have called attention in the paper also to the lateral flexibility provided by swinging links. It will be recognized that this is simply an extension of what is provided in the ordinary platform spring-suspension. Our experience with various spring-hangings indicates that the higher the spring is above the ground, the less difficulty is experienced from the lateral vibration; that is, the very low underhung spring seems to affect the body very much more than the higher spring. But there are undoubtedly very heavy stresses put in with any height of spring, due to both the side action of the axle and the effect produced by attempting to raise one wheel higher than the other with respect to the ground. The torsional effects produced by this are very great and also the bending effort that must take place, due to the fact that the projected length between the springs when the axle is at an angle is less than the distance between spring-eyes on the frame, unless some means of accommodation is provided either on the axle or on the frame. I may say that the use of some arrangement of this type on front springs had a most noticeable effect in reducing the disagreeable rattle that we get in so many cars in the front springs. I take no credit for finding out this point in spring-suspension. It was called to my attention very emphatically by a chauffeur of one of our early owners. I did not believe him when he told me, but he promptly convinced me by rocking the car back and forth and allowing me to put my hand on the shackle. We found there was from 0.003 to 0.005-in. play in the shackle and yet the rattling from it was very unpleasant.

W. C. KEYS:—What types of universal-joint and propeller-shaft were used and which types have been found best? Also, has Mr. Crane made a really fair comparison between this suspension and a very flexible full-elliptic suspension that is used on two or three cars with which we are familiar and that gives fore-and-aft and side flexibility?

H. W. ALDEN:—There are two features of this spring-suspension that I wish to question Mr. Crane about. We used a somewhat similar construction in 1905 or 1906 in the Columbia car; it was not exactly the same but it had some of the elements. We had a radial arm on each end of the axle and a link running from there to the frame. The springs were overhung. The forward end of the spring was rigidly attached to the frame at a point some 4 or 5 in. below the point of attachment of the side-rod. What we were striving for was to get a sort of pantagraph action, so that the pinion shaft would remain substantially in a horizontal position all the time; in other words, parallel with the transmission shaft. We found that we got a rather more vertical whip of the universal-joint at the rear end than we did with the straight Hotchkiss-drive. Also, we developed another very serious defect that we tried to overcome, although it never was overcome satisfactorily. It was the varying angular positions of the two vertical arms on the ends of the axle, when the plane of the chassis departed from a position parallel to the central line of the rear axle; in other words, when the car rolled sidewise. Our side-rods were very nearly in horizontal and vertical positions. In the construction that Mr. Crane uses, the side-rods are at a considerable angle. It would seem that when the car rolls sidewise the side-rods in Mr. Crane's arrangement would exert a very considerable torsional effect on the body of the axle housing. In other words, if the car rolls to the left, the lower end of the left-side axle-bracket would be pushed back and the one at the other side would be pushed ahead considerably. We were continually shearing off those brackets on the ends of the axle. I would like to know whether Mr. Crane has had any of this same experience. It seems that the strains would be terrific at times, because the coupling is absolutely rigid and something would have to give.

I notice that one of the reasons Mr. Crane favors this construction is that it decreases the vertical whip of the rear universal-joint. It would seem from the diagram that, with the chance for the rear axle to surge forward and backward and with the two lower ends of the arm rigidly held with respect to the frame, this surging action would introduce a very considerable vertical whip of the universal-joint at the forward end of the rear pinion shaft.

G. W. CRAVENS:—In 1917, when I designed the Elcar, the question of the Hotchkiss drive came up and we finally adopted it; the car is now in its fifth season and no change has been made.

In Fig. 1 of Mr. Crane's paper, I notice that he has made the rear end of the spring higher than the front. In riding in a car, the discomfort of the passengers is greatest when the body has a front-and-back movement or tendency to change its rate of speed while traveling; in other words, if the body moves along smoothly at a uniform rate and almost without swing, it is not uncomfortable. When the wheels strike obstructions they tend to stop momentarily; then, through the rebound, they come back, which means that they are tending to move ahead with relation to the travel of the body. If the rear end of the spring is lower than the front, the bound of the axle or the tendency to jump backward does not pull the body back, as is the case when the compression of the spring makes it necessary for the axle to move forward, and that inevitably tends to retard the movement of the body. The question presents itself: Is the excessive flexibility, due to having everything on shackles, provided to overcome that condition; and, does it do so?

How does Mr. Crane feel in regard to using the propeller-shaft tube as the torque-arm and mounting the springs on the axle by spherical bearings or joints like those of the Lanchester car, so that the axle is free to move and the spring is free to move on it, with the torque or push of the rear axle transmitted to the body through a spherical joint at the front end, similar to that used on the Lafayette and some of the other passenger cars built in this Country?

W. W. WELLS:—It seems to me that the angularity of the drive-rod is an important consideration. When I read Mr. Crane's paper I immediately wondered what would happen if on our 1-ton truck we mounted the rear end of the drive-rod 9 in. from the center of the axle. We have a 550-lb. spring, and I find that a horizontal force of 9200 lb. applied to the axle would move it back 1 in.; that is, the change in location of the drive-rod would be equivalent to introducing a spring of 9200 lb. per in. in the drive-rod. I notice also in Fig. 1 of Mr. Crane's paper that the drive-rod stands at an angle of about 1 to 8, which means that an upward movement of the axle of 1 in. would cause a backward movement of ⅛ in. Comparing these results shows that it would take an upward thrust of 1100 lb. to compress the spring 2 in. and shove the axle back ¼ in., due to the action of the drive-rod. A horizontal force of 1300 lb. would be required to rock the axle back ¼ in.; that is, if the direction of the blow on the axle is 45 deg., the backward movement of the axle in relation to the frame is due more largely to the vertical component of the force and the angularity of the radius-rod than to the horizontal component acting through the rocking action of the axle. Are any definite data available as to the correct angularity of the drive-rod or of the front spring?

MR. CRANE:—Answering Mr. Keys' question, the type of universal-joint used in the shaft of this arrangement at the rear end is the jaw type; that is, it has long, parallel jaws, in which rollers operate. There are two rollers in dumb-bell form on the ends of the propeller-shaft which are roller-bearing supported and the combination provides almost unlimited angularity with a very great freedom of longitudinal movement; the last being very desirable to keep excessive strains from being placed on the ball bearings either in the rear-axle pinion-shaft or in the gearbox.

I am very glad Mr. Keys spoke of the full-elliptic spring. I thought that I had done pretty full justice to this type of spring in my paper. There is no question that it is an excellent type for absorbing road shocks. The chief difficulty with it is that it is not satisfactory for absorbing torque reaction. This is due to taking it on only one-half of the spring-steel instead of all of it; in other words using the lower half of the full-elliptic steel spring to take the torque reaction as against the full spring in the semi-elliptic. I think that point is covered in my paper.

Replying to Mr. Alden's questions, I think an inspection of this device will indicate a complete dissimilarity from the one that he describes. I remember that old pantagraph arrangement very well. We considered it at the time but finally decided to use a modified Hotchkiss-drive instead. We discarded the Columbia device for exactly the reasons Mr. Alden gave in connection with it; that is, the extreme rigidity of the two sets of arms made it impossible for the axle to take an angular position with relation to the frame, or required that something should spring very considerably to allow the axle to assume such a position.

The fundamental difference in the two devices is that in the original arrangement described by Mr. Alden the front end of the semi-elliptic spring was rigidly attached to the frame, and in this device both ends of the semi-elliptic spring are shackled to the frame with relatively long shackles. There is, therefore, in the latter no very great strain set up by the axle taking an angular position with regard to the frame. Any such strain that is set up is simply spring-reacted, or partly taken care of by the swinging side-links. As a matter of fact, the car on which this was first used had the lightest rear-axle construction that I have known of being placed under a similar car. It had an aluminum center casting, carrying the gears, with axle tubes bolted to the casting on flanges of relatively small diameter. The tubes were parallel and simply flanged at the ends and, according to my recollection, they were not over 2½ in. in diameter as regards the tube and not over 5 in. as regards the flange. That car weighed 4700 lb. without passengers and developed fairly high speed under the conditions in which tests were made, having an engine of from 80 to 90 hp. It never exhibited the slightest signs of distress in any of the axle parts; due to the loading from this device, the tubes did not loosen on the axle housing or show any tendency to do so, and they were not keyed but were simply fastened with fairly well-fitted bolts. I am well satisfied that, due to the spring reaction opposing such strains, there is no deleterious action produced in the axle itself and no excessively heavy construction is required to take care of such action.

As to the vertical whip of the universal-joint, in the ordinary Hotchkiss drive this is due to the straight torque-reaction; that is, if the brake is put on sharply, the front end of the pinion shaft ducks; if power is put on violently, the front end rises. Due to the fact that we reduce the torque reaction in this device, that is, absorb it directly by a change in the angular position of the axle, the vertical whip of the end of the pinion shaft is very greatly reduced. The difference is very noticeable. This device was substituted on a car for a typical Hotchkiss drive; then we replaced the device with the original Hotchkiss drive, so that we could measure up very closely the relative action of the two arrangements.

In response to Mr. Cravens' question regarding the position of the rear springs, I realize that in a regular Hotchkiss drive it is not possible to place springs in this position, due to the fact that we depend on the position of the spring to make the axle travel in the correct relation to the frame as the springs are compressed. In this arrangement, however, the distance-rod controls this feature to an almost complete extent, and we therefore in this particular job took advantage of what we felt we would find by lowering the front end of the rear spring to keep it entirely clear of the floor-boards. The result is that a body placed on the chassis illustrated in my paper shows an entire absence of obstruction in the floor from any part of the spring-suspension. There is nothing that cuts the floor at all except the ordinary wheel housings for the fenders. I am not absolutely convinced that this is the best arrangement as shown, but in all preliminary tests it has functioned fully as well as we expected.

I also explain in my paper the reason for overhanging the rear-spring, which made this tilted position necessary; it is a fact that the higher the springs are placed, the less rolling action there is. It is conceivable that if we mounted a sort of gallows-frame construction on a rear axle and hung the body on that, and located the springs above the body, the latter would actually swing out on a curve, rather than the opposite. There is no reason it should not do that, if it were hung at the top instead of supported at the bottom.

Mr. Cravens brings up the question of the torque-arm drive, which is a very common European practice. The Rolls-Royce has used it for a number of years; as he said, it is incorporated in the Lanchester and several other cars; it is employed also on the Buick, the Lincoln and the Lafayette cars in this Country. My objection to this construction is based largely on its excessive rigidity in driving, in torque reaction. If the torque-arm is long enough, there probably is no very serious trouble from the lack of parallelism between the rear-axle pinion and the gearbox and engine shafts. If the torque-arm is short or the springs are very soft, there is a great amount of this trouble.

I overhauled one of the old four-cylinder Fiat cars after it had had about 18 months of use in this Country, a number of years ago. There was not a single joint in the driving system of that car that was not shot to pieces; every key was loose in its keyway, and the cost of bringing that car back to its original condition was excessive. Further, it was a perfectly futile expense because it was an absolute certainty that the car would again relapse into a condition of looseness as soon as it was driven for any length of time on our rough roads.

I was told by a man connected with the American Locomotive Co. that they even had difficulty with the loosening of flywheel bolts on the old Alco car that used a very stiff torque-arm construction. It seems obvious to me that, in a job where we want cheaper service, the softer we can make all reactions, the less danger we have of shaking something loose, either by road conditions or by rather rough operation on the part of careless drivers. I would say also that my chief objection to the cantilever spring is that it requires the use of a torque-arm construction. On the fairly smooth roads in Europe, that is not important; or if you use a very stiff rear spring, it is not particularly important. There is another practical disadvantage in the cantilever spring that is one of my pet aversions; it is the difficulty of mounting bodies with that arrangement. The overhang from the last point of support of the spring-suspension in an ordinary cantilever spring toward the rear of the car is extremely great. I have been dealing myself with large cars, and the weight figures are something of this order: We have two tires and a tire carrier at the extreme back of the chassis frame something like 5 ft. from where a cantilever-spring pivot would be placed. These tires and tire carriers together weigh more than 150 lb. We have a 24-gal. gasoline-tank that, with all its fittings, probably weighs 200 lb. This is not so far back as the tire carrier, but it is far enough. In addition, on a long body, there are three passengers on the rear seat who are nearly as far back as the tank; they may weigh from 500 to 600 lb. Due to the position of the rear door in any type of body, and because of other considerations, it is a very difficult matter to make the frame stiff enough to carry this load properly. It can be made strong enough, there is no difficulty about that, but all of us who have had much work to do with closed-body mountings on the chassis realize that strength in a body frame on a passenger car does not mean anything; stiffness is the one necessary feature.

Mr. Wells brings up the question of the angularity of the distance-rods shown in Fig. 1 of my paper. We have not made any very considerable tests on this particular layout by varying the angularity. What we have attempted to do is to cause the ball end of the distance-rod attached to the axle to rise and fall substantially in a vertical direction with respect to the axle. Of course, there is always a radius due to length of the distance-rod, but the radius is approximately normal to the axle. This drawing may be a little misleading, due to the fact that it is shown with the spring not very heavily loaded. As an actual fact, the fully bottomed effect occurs when the axle tube, which can be seen in Fig. 1 with a rather small diameter, bottoms on the frame. If you will trace the motion of the distance-rods, you will find it is not exactly a vertical one; it is inclined slightly to the rear, about the same amount that we would incline a Hotchkiss-drive spring attachment of the normal type.

There is no question that the angle of tilted springs, where the rear end is lower than the front end, does help considerably; but it is evidently only a compromise that possibly will be correct for a given loading and speed of the chassis and a few other conditions. It is better than nothing; but, according to our experience, it does not go nearly so far in ameliorating the difficult conditions that we have to meet as the arrangement shown in Fig. 1.

TEMPERATURES OF PNEUMATIC TRUCK-TIRES[1]

By F O Ellenwood[2]

After pointing out that the operating temperature is a vital factor in the life of a pneumatic truck-tire, the author outlines an investigation that was conducted at the plant of the Goodyear Tire & Rubber Co. This sought to determine (*a*) the best means of measing tire temperatures; (*b*) the temperature effect of inflation-pressure, load, long runs, frequency of stops, and the sizes of the rim and the tire; (*c*) the temperature of various designs of tire; and (*d*) some suitable means of reducing large-tire temperatures. The main reason for the rise in the temperature of a tire is stated to be the generation of heat resulting from rapid flexing; and the various factors having to do with this generation of heat and its dissipation to the atmosphere are listed.

The laboratory testing-machine and the methods and apparatus employed to measure the temperatures are described. One method was to measure the initial temperature and determine the rise by the increased pressure as shown by a gage fastened to the hub. The necessary corrections for air leakage and changes in the volume of the tire during operation were determined; the results are presented graphically. A type of tire thermometer was subsequently designed and used, but this was not satisfactory on account of the great differences between the thermometer readings and the temperatures as calculated by the pressure-volume method.

Thermocouples placed inside the tubes were also tried and results that checked fairly well with those calculated by the pressure-volume method were obtained. Also a detachable type of thermocouple was placed in a hole drilled through the tread to the cushion or carcass as desired.

The necessity of knowing that the air in the tubes was not saturated required the use of an air separator when inflating the tires. The effects of convection currents inside the tube on the thermocouples and thermometers are discussed. The theory as to the manner in which the temperature of a tire is reduced by using a small quantity of water inside the tube is given and

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[1] Buffalo Section paper.

[2] Professor of heat-power engineering, Cornell University, Ithaca, N. Y.

convincing experimental evidence is presented to substantiate this theory.

After summarizing his conclusions, the author expresses the belief that the study of pneumatic-tire temperatures is of much greater commercial importance and possibly of greater scientific interest than is generally realized. The rise in the temperature of a tire is a direct and sensitive indication of the waste of energy due to its operation. While this loss cannot be eliminated, it can be reduced by proper selection and combination of materials.

The development of pneumatic tires has been very great and rapid, so that now we secure wonderful service if they are given half a chance in the way of proper care. It is only natural that owners of trucks should desire pneumatic tires on their vehicles used in service where speed is of considerable importance. These large tires carrying heavy loads have brought many new and perplexing difficulties to their manufacturers. One such problem is that of making a large tire that will carry heavy loads for long distances at continuously high speed in hot climates and yet have a long life for the casing and the tube. Under such conditions there has been some trouble from premature failure and yet the same casings and tubes would be very satisfactory in cool climates, thus showing clearly that the operating temperature is a vital factor in the life of a tire. The investigation described herein was therefore undertaken for the purpose of trying to do the following things:

(1) Find out the best means of measuring tire temperatures
(2) Determine the temperature effect of inflation-pressure, load, long runs, frequency of stops, undersized rims and tire size
(3) Determine the temperature of various designs of tire
(4) Find some suitable means of reducing large-tire temperatures

The experiments have all been made during the last 2 years at the Akron plant of the Goodyear Tire & Rubber Co. and I desire to express my appreciation of the commendable policy that permits a large part of the information thus far obtained to be given to the public.

What makes a tire become hot? This question has been asked many times by laymen and often by engineers. Various answers have been given. Many laymen think it is due to the slippage between the tread and the road.

This is a factor of only the very slightest importance in nearly all cases. The main reason for the rise in temperature of a tire is the generation of heat by the rapid flexing of it or, in other words, its hysteresis.

A tire is made up of many layers of cotton cord and rubber compound, each of which has been most carefully prepared and fitted together in a very definite and painstaking manner; then the whole mass is cured under heavy pressure at a definite temperature for a certain time. This gives a tire that is remarkable on account of its strength, wearing qualities and flexibility. Although it is probably more flexible than any other material of equal strength, it is not perfectly elastic; hence, each time a section of such a tire is bent, energy appears at that section in the form of heat. This heat is conducted to the adjacent parts, so that the tire temperature rises rapidly immediately after starting the tire, and then less rapidly to some maximum value where it remains so long as the operating conditions do not change. This maximum temperature is reached when the generation of heat is just balanced by its dissipation to the atmosphere. The first of these two factors is influenced by

- (1) Amount of Flexing
  - (*a*) Load
  - (*b*) Inflation-pressure
  - (*c*) Size of tire (chiefly sectional diameter)
  - (*d*) Supporting surface (flat, round, rough, smooth)
  - (*e*) Torque
  - (*f*) Side-thrust
- (2) Rate of Flexing
  - (*a*) Speed of vehicle
  - (*b*) Circumference of tire
- (3) Kind of Tire
  - (*a*) Size
  - (*b*) Design (thickness in particular)
  - (*c*) Fabric
  - (*d*) Compound
  - (*e*) Building
  - (*f*) Curing
  - (*g*) Previous use
- (4) Sunshine
- (5) Tread Slippage

The rate at which the heat produced is dissipated is regulated by

- (1) Atmospheric Conditions
  - (*a*) Temperature

  - (*b*) Wind
  - (*c*) Moisture
- (2) Speed
- (3) Stops
  - (*a*) Frequency
  - (*b*) Duration
- (4) Supporting Surface
  - (*a*) Temperature
  - (*b*) Material
- (5) Mounting on Vehicle
  - (*a*) Mud-guards
  - (*b*) Nearness to exhaust
  - (*c*) Kind of wheel
- (6) Kind of Tire
  - (*a*) Size
  - (*b*) Thickness
  - (*c*) Material
  - (*d*) Previous use
  - (*e*) Nature of external surface
- (7) Artificial Cooling

Some of these factors are of only minor importance, while others are vital. Those of the utmost importance are

- (1) Speed
- (2) Load
- (3) Inflation-pressure
- (4) Length of run
- (5) Kind of tire
- (6) Atmospheric temperature
- (7) Sunshine
- (8) Supporting surface

A glance at these factors will indicate that it is not always easy to answer the common question, How hot does a tire become? Furthermore, a little consideration given to the question, What is the best way to determine this temperature? will show that this is a matter that cannot be answered easily. A search of the scientific literature yielded almost no information concerning the temperatures attained by pneumatic tires or the methods of measuring these temperatures. It was therefore necessary to develop methods of making such measurements.

From the beginning of the experiments it was felt that two or more independent methods of determining the tire temperatures should be used, if possible, to have a check on the results. One method that appeared to be accurate and reliable was the pressure-volume method or, in other words, measuring the initial temperature and the increase of the air pressure and volume very accurately and then calculating the temperature of this air in the tire. The question of how to determine the pressure was next in order. A detachable-gage method could not well be used as the leakage of air each time the connection was made and broken would be too serious. It was therefore necessary to use a stationary gage with some form of special connection to the rotating wheel, or mount the gage upon the hub of the machine so that it would turn with the tire. This last method was used very successfully. The wear on the gages is appreciable but not nearly so much as had been feared, especially by the gage manufacturers. Some gages are in fairly good shape after having been turned over more than 6,000,000 times.

Fig. 1 shows a tire and its attached gage mounted on the testing machine. The gages were taken off and calibrated frequently by the usual dead-weight tester. They were also checked in place at the beginning of each run by the large portable standard-gage shown at the right of the illustration. This check was made also for the reason that it is essential that the initial pressure be

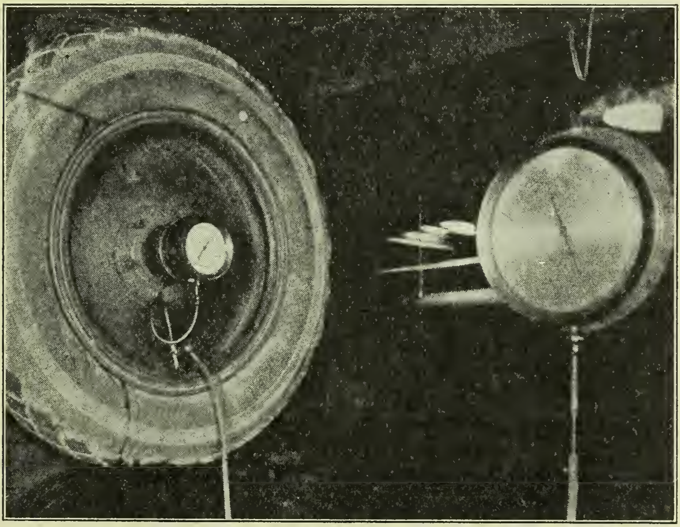

FIG. 1—GENERAL VIEW OF THE 40 x 8-IN. TIRE MOUNTED ON TESTING MACHINE SHOWING THE PRESSURE GAGE FASTENED TO THE HUB OF THE WHEEL AND THE LARGE STANDARD TEST-GAGE CONNECTED TO THE TIRE FOR DAILY CALIBRATION OF THE PRESSURE GAGE

obtained as accurately as possible, since all subsequent calculations are based upon it. To connect this large portable standard-gage with the tire under test it was necessary to design and build a special valve-opener so that the connection might first be made to the valve-stem and then the valve itself opened without losing any air. With proper attention paid to the gasket in this special valve-opener it proved very satisfactory.

Fig. 1 also shows how the tires were mounted for these tests. The cast-iron pulleys by which the tires are driven are about 45 in. in diameter and a suitable framework holds the tires in position on top of these pulleys. A known load is applied by weights to the axle on which the wheels are mounted.

LEAKAGE CORRECTIONS

There is always bound to be more or less leakage from any inner tube. This leakage depends upon a number of factors, such as the composition, thickness, temperature, age, and previous use of the tube, and the pressure of the air inside it. It is therefore necessary that the leakage shall be determined carefully for each run, thus giving the leakage under the complex conditions just as they exist.

The best way to make the leakage correction is to determine the average rate for the 24-hr. period following the start of the test. If this rate is small, as it usually is, the distribution of it may very properly be made on the basis of time alone. If it is appreciable the additional factors of temperature and pressure should be introduced in its distribution. Special effort was made to secure correct leakage corrections in all of these tests. A very close check was obtained on the observed leakage of one old tube by taking the pressure readings each hour during a 6-hr. run and the 18 hr. following it, then calculating the leakage on the assumption that it would be proportioned to the absolute temperature of the air in the tube and to the square root of the gage pressure. This method of distribution proved to be very satisfactory. Special curves were prepared to facilitate its application. In obtaining the 24-hr. leakage rate it will be found that usually the temperature and volume of the air in the tire are not the same at the end of this period as they were at the beginning. This necessitates that these values shall be determined and the proper corrections applied.

VOLUME CHANGES DURING OPERATION

Soon after beginning this work it was found that a pneumatic tire changes its volume very appreciably with variations in pressure, temperature and use. Various means of measuring this expansion were tried, the most convenient and accurate one being the measurement of the longitudinal tread circumference by a steel tape. After the increase in tread circumference has been determined, calculations may then be made to show the corresponding change in the volume of any tire whose cross-sectional dimensions are known. Fig. 2 shows such relations for three different sizes of tire. Fig. 3 shows directly the temperature corrections that are due to

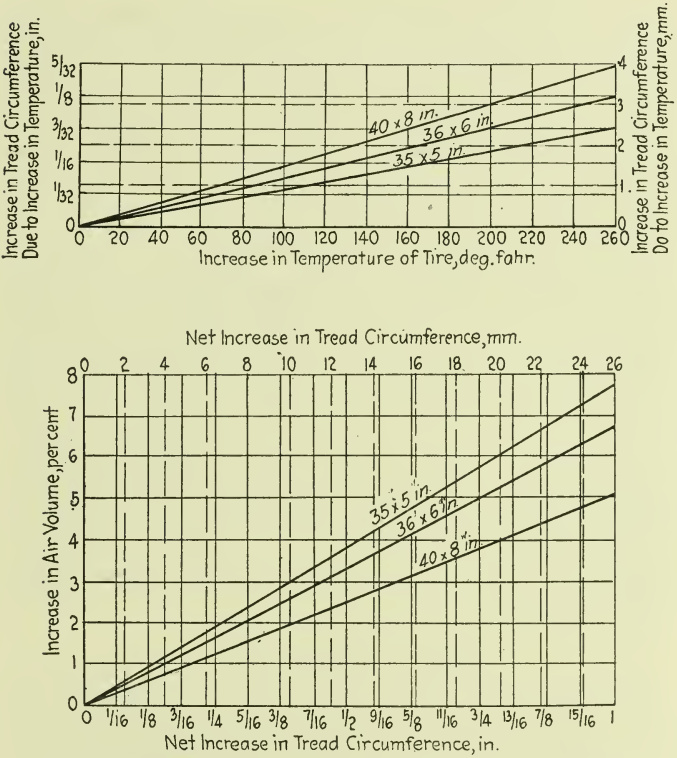

FIG. 2—CURVES SHOWING VOLUME CHANGES IN OPERATION

The Upper Curves Show the Increase in the Circumference of the Tread Due to the Thermal Expansion of the Tread and the Carcass, while the Lower Set Shows the Percentage of Increase in the Air Volume for Various Increases in the Tread Circumference Resulting from the Stretching of the Tire changes in tire volume, for various initial temperatures and pressures. The temperature corrections for the change in volume are surprisingly large for the first day's run, being as much as 15 deg. fahr. for a 40 x 8-in. tire in some cases, while the corrections on subsequent days would be only from 3 to 6 deg. It is also very interesting to note that after the tire has been run for a considerable mileage and then deflated and left to stand for an appreciable time, it will also nearly recover its original volume. When a tire of this kind is again used its volume will increase very materially during its first run after its long rest, just as a new tire would do.

TIRE THERMOMETERS

Soon after attempting to ascertain the best ways to measure tire temperatures, we believed that a successful tire thermometer might be designed and built. There would be two advantages in such a thermometer. In the first place, if it could be built to stand up under operating conditions and at the same time be reliable and not too much trouble to install, it would be the means used in all future tests to determine tire temperatures. On the other hand, if it were only partly successful it certainly would be worth while to have one solely for the purpose of obtaining the actual temperature of the air in the tire at the beginning of each test. This initial temperature is very important because all of the subsequent temperatures depend upon it when calculated from the pressure and volume measurements. The only other way of determining this initial temperature of the air in the tire is by placing a thermocouple inside the tube; or by keeping a record of the room temperature for several hours preceding the test and determining the lag of the temperature of the air in the tire with reference to room temperature. At the time mentioned we had not been able to make a satisfactory thermocouple to place inside the tube.

TUBE THERMOCOUPLES

We tried various ways before getting the thermocouple wires inside the tube so that there would be no leakage, the installation correct electrically, the construction strong, the couple held near the center of the air-space inside the tube, and the couples attached to the regular valve-stem. This last item is of considerable importance since it means the elimination of the trouble and annoy-

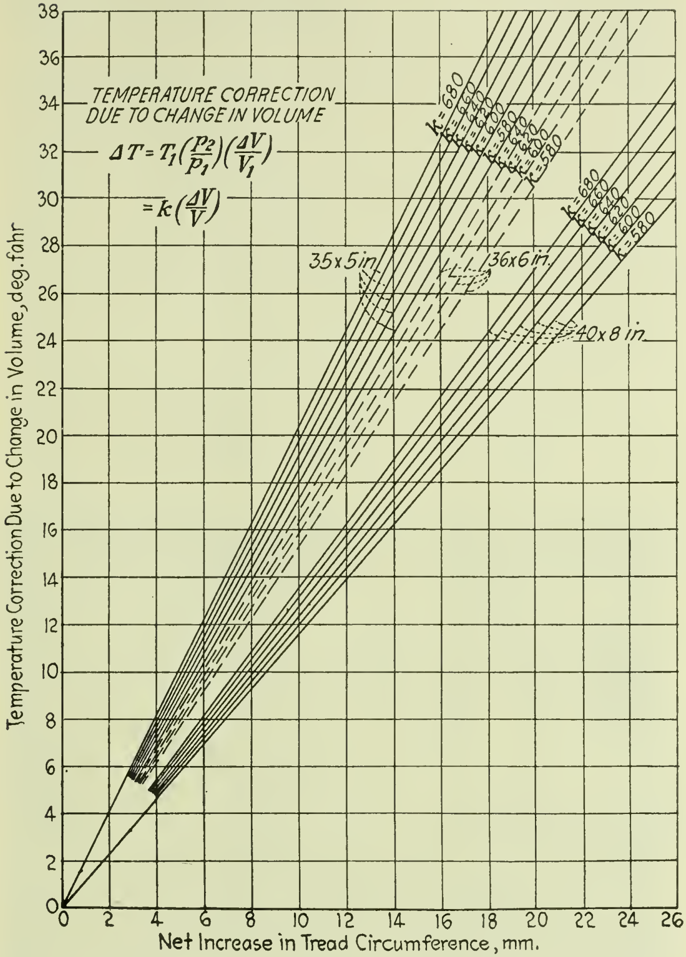

FIG. 3—CURVES SHOWING CORRECTION TO BE ADDED TO THE TEMPERATURES AS CALCULATED FROM THE INCREASE IN PRESSURE

V_1 = Initial Volume of the Air in the Tube
ΔV = Increase in Volume of the Air in the Tube
P_1 = Initial Absolute Pressure of the Air in the Tube
P_2 = Final Absolute Pressure of the Air in the Tube
T_1 = Initial Absolute Temperature of the Air in the Tube
ΔT = Temperature Correction

ance of having to prepare a special tube with its extra valve-patch and the extra hole through the rim, as well as the additional trouble of mounting the tire and tube having this extra valve-stem.

Fig. 4 is an X-ray photograph of a 40 x 8-in. tube containing two thermocouples, one near the center of the tube at *A* and one near the bottom or rim at *B*. This type is very rugged mechanically, as shown by the fact that each one survived the splicing operation without distortion, but the heat that may be conducted from the couple itself by having the couple wires wrapped to the insulated supporting piano wire is a possible objection to this type. Fig. 5 is another side-view of an X-ray photograph of a tube which was taken after the couples in this 36 x 6-in. tube had been run during many tests. The couple near the center *C* is supported by an invisible cord that runs horizontally from one side of the wrapped supporting wire to the other. The bottom couple *D* has been broken from its supporting cord. The coils of fine thermocouple

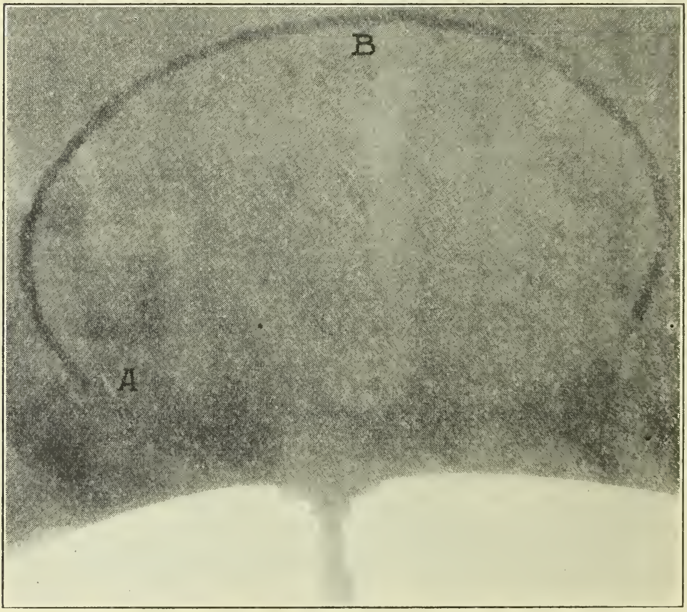

Fig. 4—X-Ray Photograph of a Side View of Part of a 40 x 8-In. Inner Tube with Thermocouples at *A* and *B*, Each Kept in Position by Wrappings of Piano Wire That Are Soldered to the Stem

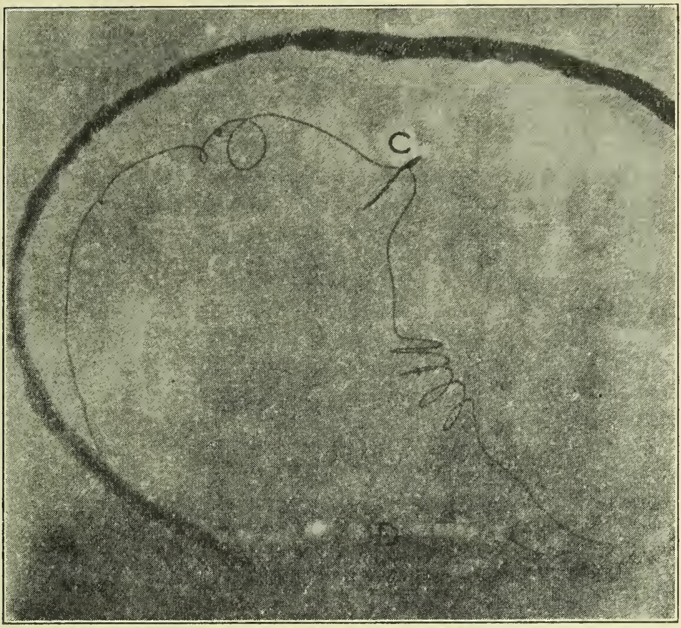

Fig. 5—X-Ray Photograph Showing a Side View of Part of a 36 x 6-In. Inner Tube with Thermocouples Held near the Center at C and the Bottom at D by Cords Attached to the Piano Wire

wire were symmetrical before the tube was spliced and inserted in the casing. However, it should be noted that despite its rough usage during splicing the center couple is still in its proper position. This is a very good type of construction as it is rugged and the heat conducted away from the couple is a minimum. With this type we obtained some very satisfactory checks on the temperature as obtained by the pressure-volume method.

Convection Currents

The low readings obtained from the tire thermometers and the tube thermocouples as compared with the temperatures calculated from the pressure and volume puzzled us very much. To eliminate the possibility of moisture getting into the tubes from the compressed air, a special separator containing calcium chloride was made and used when inflating each tube. By this means and also by samples of air drawn from the inflated tires we satisfied ourselves that the amount of moisture inside the tubes was very much less than that required for saturation. We were now sure that our calculated temperatures were correct, but we could not believe that the small wires used in the thermocouples would conduct away enough heat to cause the low readings obtained from them.

We next began the investigation of the effect of the convection currents inside a hot tube when standing still and when running. This proved to be very illuminating. We had always been reading the temperature from the inside couple when in a position some 30 deg. from the bottom (see Fig. 6) as this was the position in which the tires were always stopped to have the gage exactly right to obtain its correct reading. We now compared the couple readings in this position with those of the same couple in positions near the top to which we would quickly turn the tire. This gave a difference that usually would be 5 deg. or more. It is not easy to obtain this difference correctly as the tire must stand in one position long enough to enable the couple to acquire the temperature of the air in that position; then the time must be noted when the reading is taken to correct for the cooling that occurs after taking the reading in the first position. However, we did establish to our own satisfaction the existence of the convection currents shown by Fig. 6 for a hot tire that has just been stopped.

Using a set of collecting rings made of the same material as the couple wires themselves, we were next able to determine while the tire was running the difference in temperature readings between the couple near the rim and the one near the center of the tire. This difference was found to be large as will be shown later. The rotation of the tire causes very rapid convection currents in the transverse section of the tire. This also will be discussed more fully later. We were convinced that the thermocouples are subject to appreciable correction due to convection currents, and that the magnitude of these corrections depends upon the position of the couple relative to the center of the tube, whether the tire is running or standing still. These same convection currents would account for a considerable part of the discrepancy still found to exist in the case of the tire thermometer in a well with many fins on it. This information, coupled with that concerning the heat conduction, which was such a big factor with the tire thermometer, now gave us peace of mind regarding our temperatures as obtained by the pressure-volume method, and made us feel certain that for comparing the temperatures of different designs of tire this is the best method of which we have any knowledge.

DETACHABLE THERMOCOUPLES

After having experimented with tube thermocouples and realizing fully some of the serious difficulties that cannot be overcome on account of their being inside of the tube, it was felt that it would be worth while to experiment with detachable couples. By this term we mean couples that are inserted in small holes drilled through the tread to the cushion, breaker or carcass as desired. After making satisfactory holes through the tread, it is necessary to have a couple made of very fine wire mounted on the end of a small insulating plug that

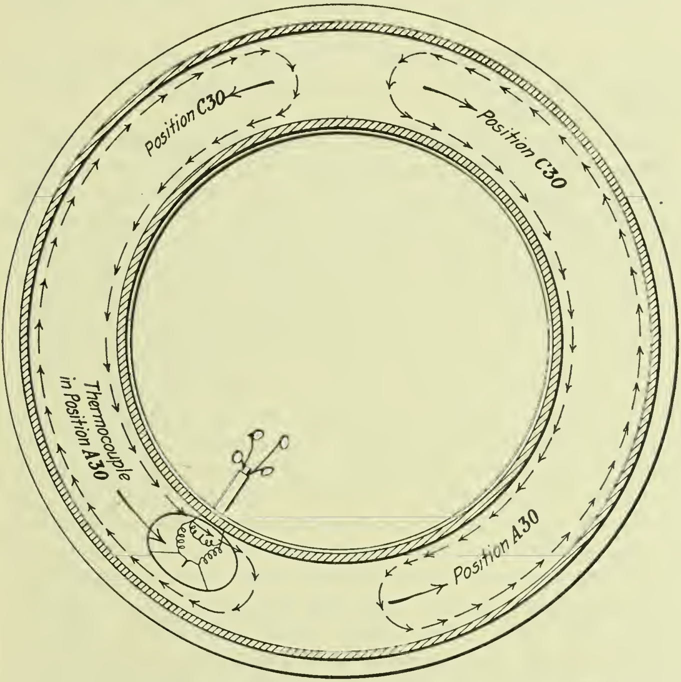

FIG. 6—CONVECTION CURRENTS IN A HOT TIRE IMMEDIATELY AFTER STOPPING

The Dotted Lines Indicate the Path and Arrowheads the Direction of the Convection Currents when the Tire Is Standing Still. All Temperatures Were Obtained by the Pressure-Volume Method with a Load of 2500 Lb. at a Speed of 30 M.P.H.

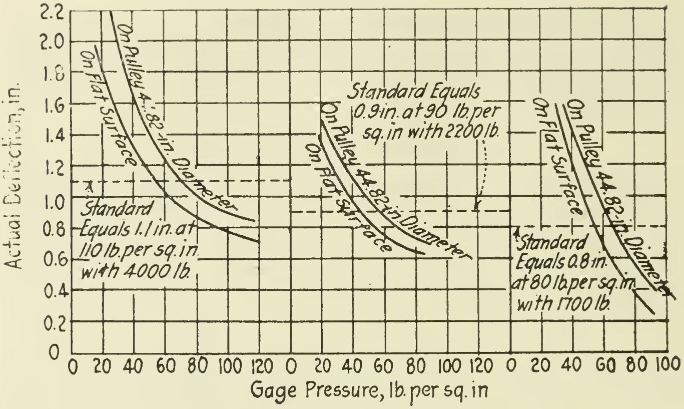

FIG. 7—DEFLECTIONS OF THREE SIZES OF TIRE HAVING VARIOUS INFLATION-PRESSURES WHEN SUPPORTED BY FLAT AND CURVED SURFACES

can be inserted in the hole. Wood and hard rubber are suitable but the latter is preferred because it is not broken so easily.

This type of detachable couple can be inserted immediately after stopping the tire and the temperature indicated by the potentiometer will rise to its maximum value very quickly after insertion. It is not, however, an instantaneous process and the reading must not be made too quickly after the insertion of a cold plug, as it may require at least 30 sec. to reach the maximum temperature. With every care that we could exercise in all particulars it was still found that for some unknown reason this method would give occasional readings which were 10 or 15 deg. too high or too low to plot a smooth curve. Such cases, however, were very infrequent and our experience indicates that this is a method of great value, especially as a supplement to the pressure-volume method. Obviously one of its greatest advantages is that it enables the temperature of the stock itself to be measured and our investigations have shown that this temperature is considerably higher than the temperature of the air inside the tube. The other chief advantages of this system are the simplicity of the apparatus and the ease of renewal of the couple. It is also obvious that it has one disadvantage, namely, the injury to the casing resulting from drilling a hole through the tread. For laboratory-test tires this in itself is not a serious matter because there is no sand or dirt to work its way inside the carcass. It seems likely that there may be an appreciable correction to be added to the readings obtained from couples inserted in a hole in the rubber or carcass. This correction is due to heat conduction through the wires from the couple, and is greater the shallower the hole. This phase of the subject is now being carefully investigated by the research division of the Goodyear company.

SOME TEST RESULTS

Before making any comparative temperature-tests it was necessary to determine the deflection of various sizes of tire when resting on the curved surface of the testing machine. By deflection of a tire we mean the vertical drop of any part of the wheel on which the tire is

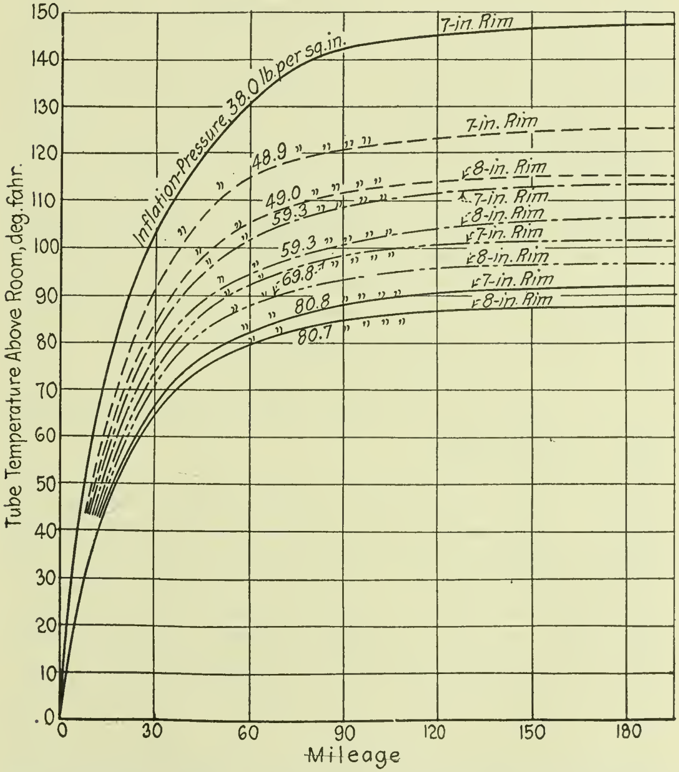

FIG. 8—TEMPERATURES OF A 40 x 8-IN. TUBE ABOVE THE ROOM FOR VARIOUS MILEAGES AND INFLATION-PRESSURES AND TWO-RIM SIZES

mounted as a result of applying a known load to the tire. This deflection causes the lower part of the tire to flatten until the contact area becomes sufficiently large to support the load with the air pressure existing in the tire.

Fig. 7 shows the results of deflection tests on three sizes of tire for various inflation-pressures, two kinds of surface and one load. These curves show what load is needed to give a normal deflection of the tire when mounted on the testing machine. For the runs that were made on the 40 x 8-in. tires this machine was not strong enough to permit full load, so the full-load deflection was obtained by reducing the inflation-pressure. This undoubtedly reduces the tire temperature below what it would be for full load and full inflation-pressure. Just what this relation may be for various sizes of tire is a matter that is still being investigated.

Fig. 8 shows several interesting things. The first is how rapidly the temperature rises during the first 60 miles, then less rapidly to its maximum, which is near 150 miles, for an 8-in. tire when running at 30 m.p.h. The smaller the tire the sooner it reaches its maximum temperature. Also observe what the temperature effect is of using an undersized rim on a 40 x 8-in. tire. This is a practice not to be encouraged in general, and one to be distinctly avoided in hot climates. It will be noticed how fast the temperature of this size of tire rises with decreasing inflation-pressures. For the load of 2500 lb. the lowest pressure, 38 lb. per sq. in., represents a deflection nearly 60 per cent more than normal. This is what many tires sometimes have to endure. These curves indicate that the "air thermometer made out of a tire" may be entitled to the high rank usually accorded this type of thermometer.

Fig. 9 shows the temperatures of the various parts of an 8-in. tire as obtained by the several methods described. The reason for the inside couple near the center of the tube giving values lower than those obtained from the pressure and the volume is probably largely due to heat conduction from the type of couple used, shown by Fig. 4, and to the convection currents. These readings from the inside couples were all obtained while the tire was running, except those taken after 5½ hr. of the test when the tire was stopped. Attention is called to the very rapid drop in temperature of the eighth ply and of the cushion in the tire immediately after stopping, at the end of 150 miles. This is a 12-ply tire with a thick

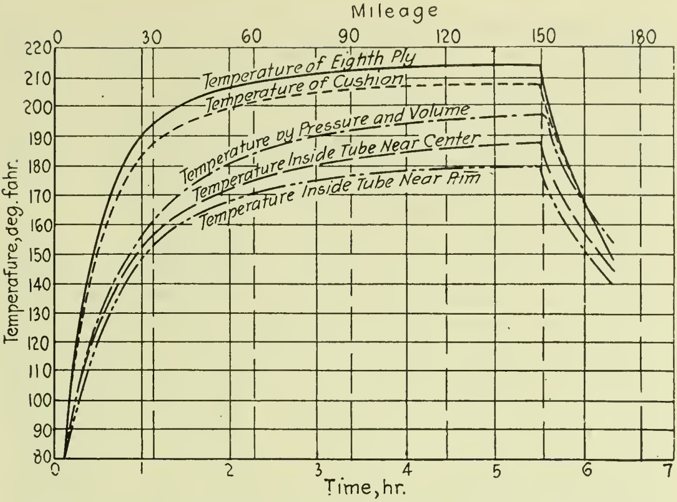

FIG. 9—TEMPERATURES OF VARIOUS PARTS OF A 40 X 8-IN. TIRE AS OBTAINED BY VARIOUS METHODS, THE ROOM TEMPERATURE IN THIS CASE BEING 90 DEG. FAHR.

tread. This test shows, as did many other similar to it, that the air in the tube is some 15 or 20 deg. cooler than parts of the casing. It also shows how quickly the high temperatures are reached by those parts of the tire in which the heat is produced, so that long distances do not need to be covered to produce temperatures that may be injurious to the carcass.

ARTIFICIAL COOLING

For those sections of the Country where there are long periods of very hot weather it is undoubtedly true that some simple means of keeping large truck-tires cooler than they are under present conditions would be very valuable. It was therefore one of the purposes of these experiments to ascertain if possible some means of doing this. The most successful method of artificial cooling investigated was that in which a small quantity of water is placed inside the tube before inflating it with air. Many such experiments, beginning in 1919, have been run and in all cases the reduction of the average tire-temperature by water has been very noticeable. The hotter the tire the greater this effect, for reasons that will be discussed later.

The theory which prompted trying water inside of the tube is this. First, we should expect a certain reduction of heat generation because of the reduced flexure of the casing by reason of the increased pressure inside of the tube due to the pressure of the water vapor. This vapor-pressure would be small but nevertheless would act as a sort of safety-valve in under-inflated tires because the hotter the tire the greater would be such pressure. The vapor-pressure rises faster than the air pressure for a given increment of temperature.

In the second place we should expect that the vapor inside of the tube would transfer a large amount of heat from the hot portions of the tube to the cooler parts of it since as the tire rotates the water is kept in a thin band extending entirely around the tread portion of the tube. A section of the top part of a rotating tire with some water in it is shown in Fig. 10. That portion of the tube covered by the water is the hottest. The water absorbs heat from the hot part of the tube and the vapor thus formed carries a large amount of heat from this part of the tube to the cooler portions lying next to the rim and sidewalls on which condensation takes place. There is a very rapid circulation of the air and vapor in all transverse sections of the tube, as indicated by Fig. 10, due to the difference in temperature between the tread and bead sections of the tire. The faster the tire rotates the more rapid is this circulation. Thus

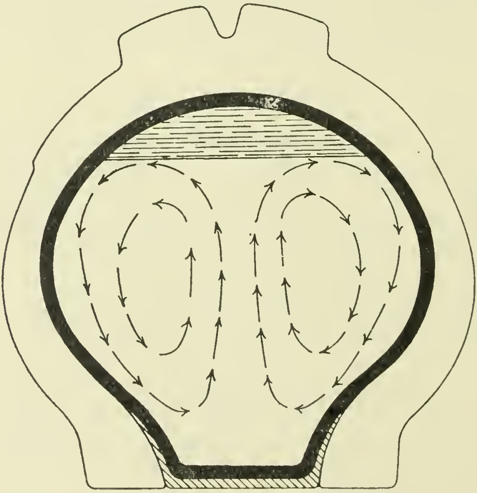

Fig. 10—Cross-Section of a Rotating Tire with about 10 Per Cent of the Tube Volume Filled with Water
The Arrows Show the Path of the Convection Currents

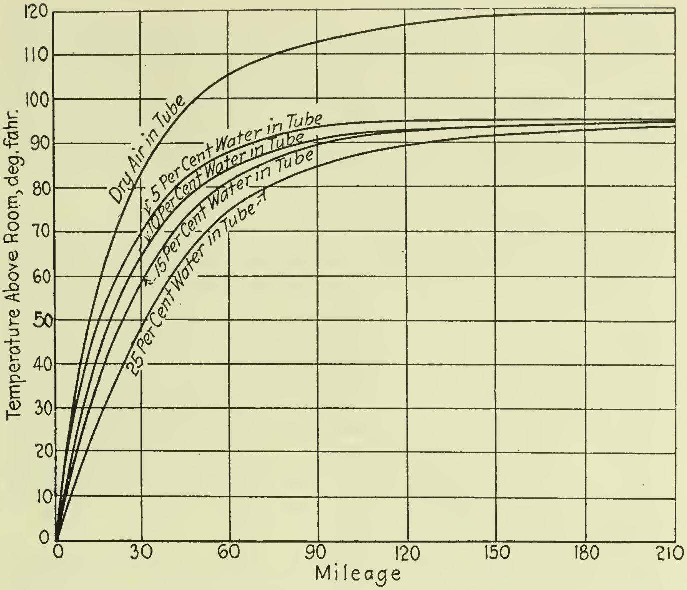

FIG. 11—TUBE TEMPERATURES RESULTING FROM VARIOUS AMOUNTS OF WATER IN THE TUBE

it is easy to conceive of an extremely rapid circulation in every transverse section of a rapidly rotating tire whether it has vapor in it or not. The addition of water-vapor produces more rapid circulation as it increases the difference in densities for equal temperature-differences. It is, however, in the vastly superior heat-carrying capacity of a wet vapor, as compared with dry air, that we should expect to find the main advantage of the vapor. The delivery of more heat to the cooler parts of the tube lying next to the rim and sidewalls means increasing the ability of the tire to dissipate heat to the atmosphere. If this be true, we should find a very marked decrease in the temperature of the hottest parts of the tube, which always fail first, and an increase in the temperature of the coolest parts of the tube, which never fail from high temperature. The average tube-temperature should also be lowered in proportion as the ability of the tire to dissipate its heat is increased. This would be best determined by the temperature of the air in the tube, as the average temperature of the tire cannot be so nearly measured by any other means.

In the third place, we should expect a cooling effect by reason of the absorption of heat by the liquid itself, until this liquid becomes heated up to the same temperature as the tube. Here we should expect a cooling effect that would depend upon the amount of liquid used and the distance run without stopping. On the other hand, the effect of the vapor will be independent of the amount of liquid used. The experimental evidence will now be presented to substantiate this theory.

Fig. 11 shows clearly the cooling effect of water inside the tube of a 40 x 8-in. tire in amounts varying from 5 to 25 per cent of the air volume of the tube or from about ¾ to 3½ gal. of water. The larger amounts are advantageous only during the first few hours of the run, as the temperatures with water in the tube all finally approach the same value. This wet-tube temperature is seen to be very appreciably lower than when using the dry tube. These curves were all taken with the same tire, tube, load and speed and also with the same inflation-pressure which was kept low to give a flexure about 30 per cent greater than normal.

Fig. 12 shows what portion of the total pressure in the tubes of two badly under-inflated tires is due to the vapor. For each tire it will be noticed that the total pressure is slightly greater with the dry tube until about 30 miles have been run. From that point on the temperature has become high enough so that the vapor-pressure more than compensates for the lower air-pressure in the wet tube due to its lower temperature. In Fig. 13 the wet tubes are seen to be about 30 deg. cooler than the dry ones after 90 miles have been run. At 30 miles this difference is about 25 deg. With these tires badly under-inflated one of the dry tubes reaches a temperature above that of the room of 147 deg. fahr. and the other one goes to 134 deg. This difference is due to the variation in the design of the two casings. With 9 per cent of water in each of these tubes in the same casings as before we have the tube temperatures reduced more than 30 deg. How much of this cooling is due to the vapor-pressure may be found from the bottom curve in each chart. This is seen to be about one-fourth of the total cooling effect, showing that the vapor must be given the credit for most of the benefit derived from the water in the tube.

The curves given at the bottom of Fig. 14 are extremely valuable as they help to explain so much that was at one time very puzzling and annoying. Reference has been

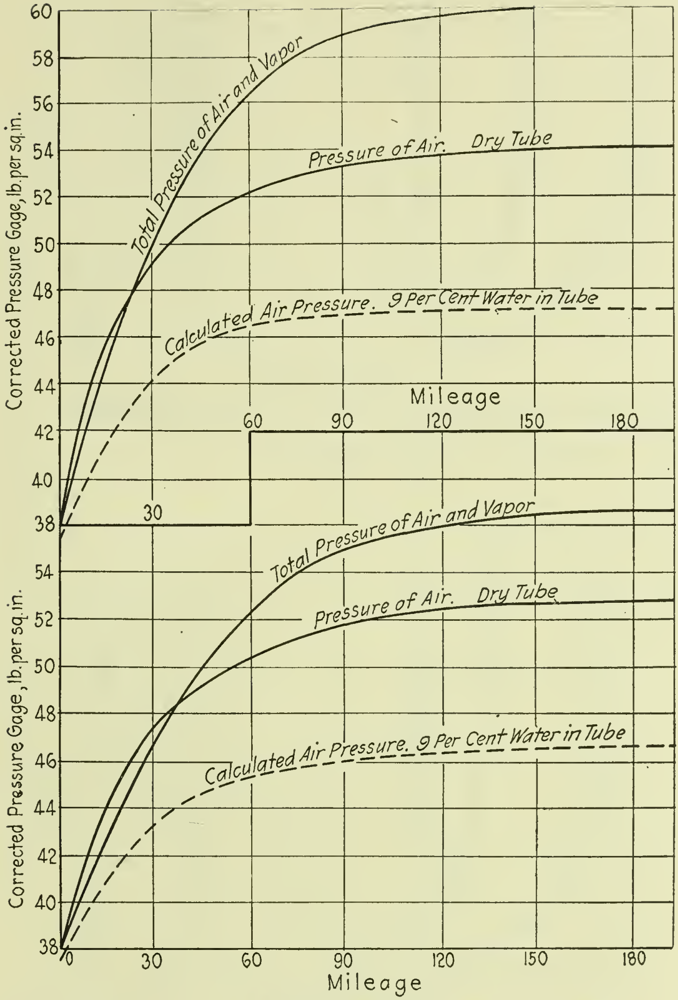

FIG. 12—COMPARATIVE PRESSURES IN THE SAME TIRES WITH AND WITHOUT WATER IN THE TUBES

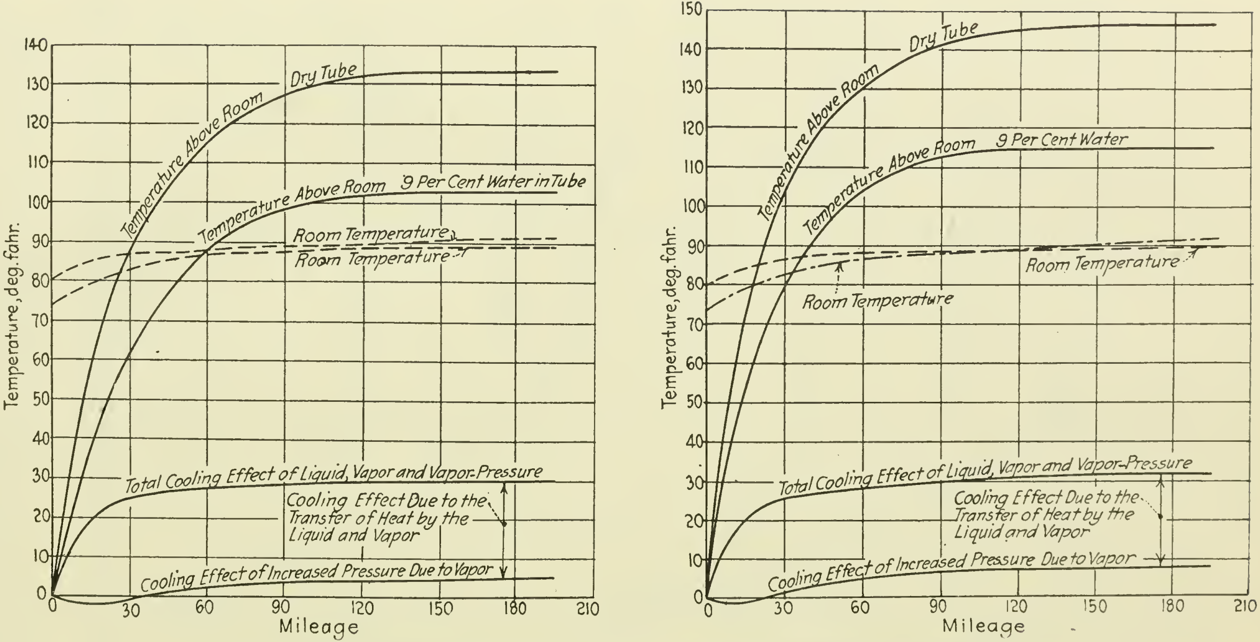

FIG. 13—CURVES GIVING AN ANALYSIS OF THE EFFECT OF WATER IN THE TUBE

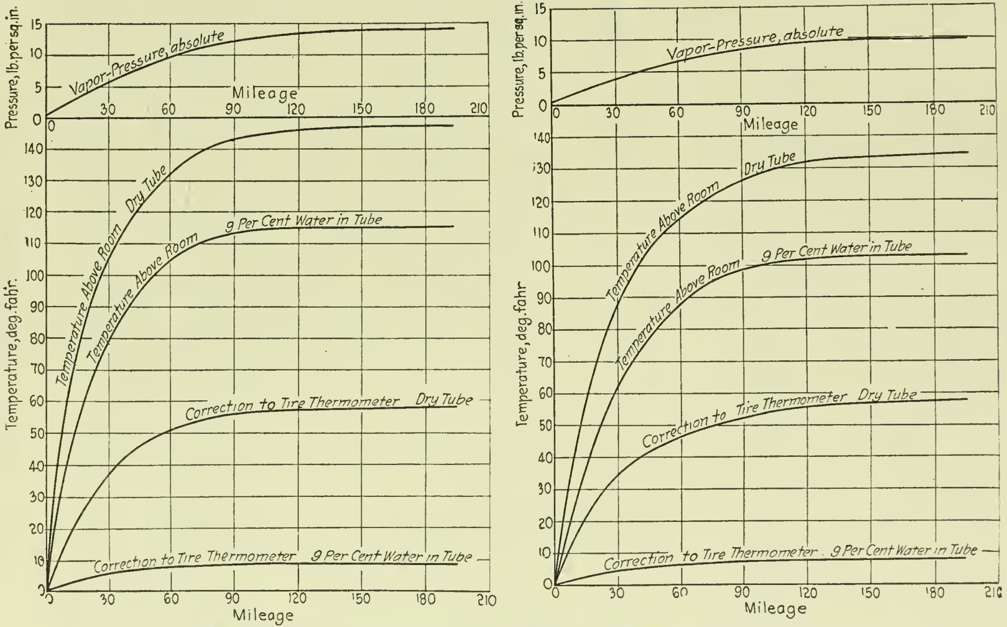

FIG. 14—CURVES SHOWING THE EFFECT OF WATER VAPOR IN REDUCING THE CORRECTION OF THE TIRE THERMOMETER

made to the fact that the tire thermometers always gave temperatures much lower than those obtained by calculation from the pressure and volume measurements. With water in the tubes the tire thermometers would for many tests give higher readings than with the dry tubes. I believed that this was because the vapor caused a greater reduction in the correction of the tire thermometer than it caused in the reduction of the temperature of the tube. To prove this was not easy. I have already mentioned the large correction of the tire thermometers due to heat conduction from the thermometer well. With wet vapor surrounding this well instead of dry air the amount of heat delivered to it would be much greater; hence we should expect a large reduction in the correction of the instrument. This made it imperative to check very carefully, as already described, the amount of moisture in the air when we were supposed to be using dry air.

In Fig. 15 may be seen the serious temperature effects of under-inflation on dry and wet tubes in a 36 x 6-in. tire. Even for this size of tire the cooling effect of the water is very marked, but is not so great as with the 8-in. tire. Just how rapidly the temperature increases with the size of tire is clearly shown by Fig. 16. This is chiefly due to the increase in the thickness of the walls of the casing as the size increases. It is obvious from this chart that ordinarily no artificial cooling is needed for tires 6 in. or less in diameter, but as we go to 8 in., and above, anything that will keep the tire cooler should be used if it is not troublesome in its application. Putting a small amount of water inside a tube is a simple matter and there is nothing further to do about it until the tire is to be used in freezing climates, when the tube should be taken out and drained. There is no change in the riding qualities of the tire by using a small amount of water. There appears to be no objection to its use and the certainty of its cooling effect on the tube makes it deserving of a trial by all truck owners who operate large pneumatic tires at high speed over long distances in hot climates.

GENERAL CONCLUSIONS

From these experiments it may be deduced that:

(1) The measurement of the increase in the air pressure of a tire is a very accurate and reliable means of determining the average tube-temperature, provided proper precautions are taken to

know whether the air is saturated and to have accurate measurements of pressure, volume changes, leakage and initial temperature

(2) Convection currents inside the hot tube are very large and may produce a large correction to any temperature-measuring apparatus that records only the particular temperature at some definite region, which is small in comparison with the whole cross-sectional area of the tube

(3) These convection currents are due to the variation in density of the air in contact with the hot and cool portions of the tube, combined with the force of gravity and that due to the rotation of the wheel. The first of these forces causes the

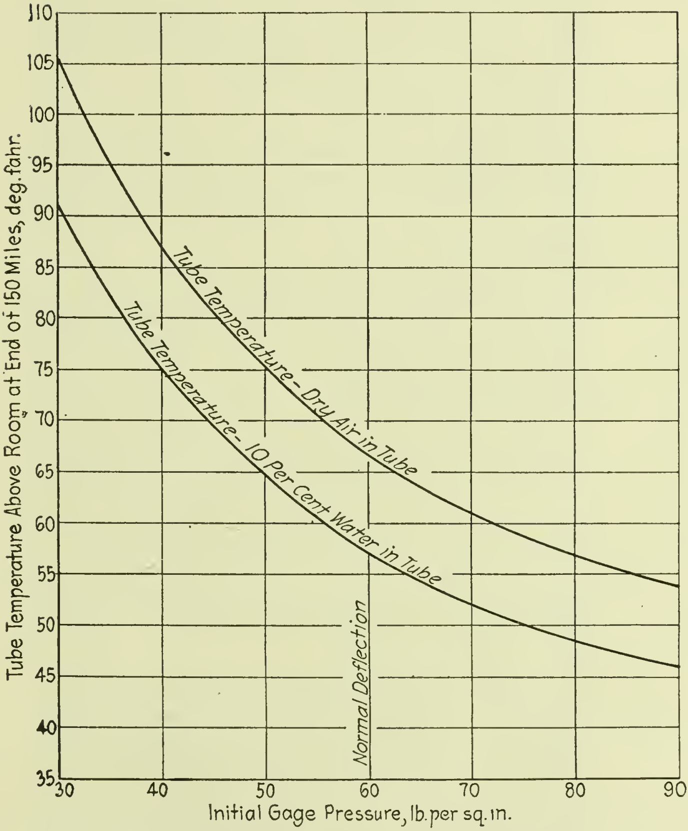

Fig. 15—Temperature Effect of Under-Inflation on a Small Truck-Tire With and Without Water in the Tube

convection currents to flow in the longitudinal direction of the tube when the tire is standing still, and the second causes much stronger currents in a transverse section of the tube when running

(4) Tire thermometers of the type described are not very well suited to any tube-temperature measurements except the initial one, on account of the corrections due to the convection currents and the still greater correction due to heat conduction to the cool part of the tube to which the well must be clamped

(5) Thermocouples placed inside the tube and mounted so that they will be held near the center of the tube are rather frail and are likely to be injured or misplaced during the splicing of the tube. They are also subject to corrections due to convection currents and require considerable care and patience to make and install

(6) Detachable thermocouples of the type described are easily made and used. They are reliable when carefully used, and possess the very great advantage of giving fairly close to the correct temperature of the various parts of the tire that are hottest and whose temperatures are often most desired

(7) The detachable-thermocouple readings must be taken at very carefully measured intervals of time after stopping the tire as the hottest parts of the tire cool appreciably and rapidly immediately after stopping

(8) It appears likely that the detachable thermocouple should have a correction applied to its reading to compensate for the conduction of heat through the wires from the couple to the outside air. This correction has not been worked out completely as yet but will be greatest when this type of couple is used in holes of shallow depth

(9) The pressure in the tube does not begin to drop until a short time after stopping the tire, thus affording time for obtaining accurate pressure and volume readings

(10) The pressure-volume method combined with the detachable thermocouple one appears at the present time to be best suited to determining pneumatic-tire temperatures, as the results are more accurate, comprehensive and easily obtained than those from any other methods thus far developed

(11) For comparative tire-temperatures the results should be given in terms of temperature above the atmosphere surrounding the tire, as this factor

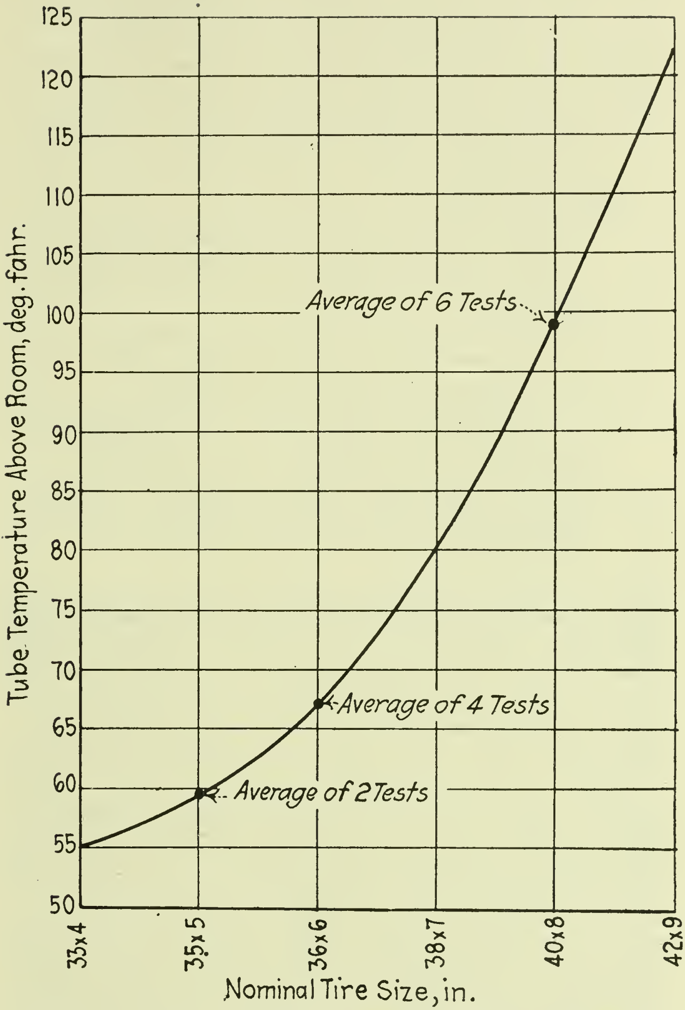

FIG. 16—VARIATION OF TIRE TEMPERATURES WITH THE SIZE OF THE TIRE

is very important and variable. It is also essential to have the same loads, inflation-pressures, speeds, distances, supporting surfaces and sunshine in order to use comparative temperatures of different designs of the same size of tire

(12) The temperature of a tire increases very rapidly with its size due to the thickness of its walls

(13) High temperatures are destructive to tires but the relation between the life of a tire and its temperature for various compounds has not yet been established

(14) Small amounts of water, from 5 to 10 per cent of the air volume, used in the tube with the air result in a cooler tire, the difference in the temperature being marked in the case of a large hot truck-tire

(15) The use of water in tires of passenger-car size is not recommended, since their temperatures are not usually sufficiently high to justify it

(16) Care must be exercised when using water in the tube to drain the water from it before exposing it to temperatures much below the freezing-point

(17) All users of tires should note the rapid increase in tire temperature with the decreased inflation-pressure, so that they may prolong the life of their tires

In conclusion, I desire to express the opinion that the study of pneumatic-tire temperatures is of much greater commercial importance and possibly of more scientific interest than is generally realized. A quick temperature-test of a tire may yield much more information than a long expensive test to destruction of the same tire. A temperature test is not, however, complete in itself for all tire testing, since tread wear, for example, requires another type of test, which, incidentally, can also probably be made better in the laboratory than on the road. Even after the correct temperatures of a tire shall have been determined, there is much research work yet to be done in order that the full significance of such information may be appreciated. This much, however, is sure: the rise in temperature of a tire is a direct and sensitive indication of the waste of energy due to its operation. The greater this loss of energy the faster the tire wears itself out and the more power is wasted to operate it. This loss can never be reduced to zero, but it can be kept low by selecting the most suitable materials and then very skilfully combining them according to the best design.

THE DISCUSSION

Ellwood B. Spear:—Only those who have endeavored to measure accurately the temperature of a pneumatic or a solid tire can appreciate the difficulties encountered by Professor Ellenwood. He is certainly to be congratulated on his success in his painstaking and important investigation. Tire manufacturers are making an intelligent and well-organized effort to give the consumer better value for his money. They will inevitably fail to get the best results, however, unless the consumer is ready to cooperate by taking proper care of his tires. I believe that one of the best methods of obtaining this cooperation is to demonstrate to the automotive engineer, the tire repairer and the consumer what happens to a tire in cases of overloading, under-inflation and fast driving. The determination of the temperature in the different portions of a tire offers a convenient means for the investigation of the effect of the design, as well as of the part played by the above-mentioned factors.

John P. Coe:—Engineers in the rubber industry are continually confronted with difficulties in scientific work due to the fact that the materials they are using are in a sense intangible compared with most of the materials used in the construction of automotive machines. Professor Ellenwood's work is all the more significant to tire engineers on that account. I have been very much interested in the data he has presented, as well as in the methods he has explained for measuring the temperatures developed, on account of the fact that we have experimented along very similar lines. Our research has not been so complete as Professor Ellenwood's, however, and is covered in most respects by his report. We use a slightly different arrangement for measuring the temperatures of the tire wall in that we attach our thermocouples to an awl point and are able thereby to insert them quickly into any part of the tire desired. We use small wire for thermocouple leads and solder the couple to the point of the awl. This outfit does not last indefinitely but is more durable than might be expected.

It has been very interesting to hear Professor Ellenwood's discussion of the effect of adding water to the air within the tube, inasmuch as we made this discovery ourselves without previous knowledge of his work. It is, however, evident from his report that he antedates us in the discovery.

PRINCIPLES OF MOTORBUS DESIGN AND OPERATION[1]

By G A Green[2]

In the paper an attempt is made to answer the broader phases of the questions: What constitutes a bus? and In what respects does a bus differ from other classes of automotive equipment? by establishing the principles on which the design and operation of motorbuses should be based. The treatment of the subject is in the main impersonal, although specific references to the practice of the Fifth Avenue Coach Co. and illustrations of its equipment are made to emphasize the points brought out. The questions of the unwisdom of overloading, rates of fare and the service requirements are discussed briefly as a preface to the paper proper.

The factors controlling bus design are stated to be (*a*) safety, (*b*) comfort and convenience of the public and (*c*) minimum operating cost. The various subdivisions of each are commented on in some detail, and numerous illustrations and tabular data supplement the text. The conclusions reached are that trucks or automobiles, either modified or unmodified, are absolutely incapable of rendering satisfactory and economical service as buses; such failures of buses as have occurred were due to the combination of extemporized equipment, indiscriminate operation, overloading and lack of experience; and, if the Society would concentrate its standardization work on the motorbus, much good could be accomplished.

The questions that builders and intending operators are asking today are, What constitutes a bus? and In what respects does a bus differ from other classes of automotive equipment? There seems to be a general agreement that a properly designed bus has special requirements; that it differs materially from equipment such as trucks and automobiles.

I have been requested to give the Fifth Avenue Coach Co.'s views on this subject. It is, of course, possible to deal with only the broader phases. No attempt will be made to discuss detail design, but merely to establish the principles on which it is thought such design should be based. We believe that with problems of this character, it is principles that really count, that once having clearly established them, the rest is comparatively easy. Actually, there is no real mystery in motorbus design. It is purely an engineering problem and there is available ample engineering talent to afford its solution, but the principles must first be established.

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[1] Semi-Annual Meeting paper.

[2] M.S.A.E.—Vice-President and general manager, Fifth Avenue Coach Co., New York City.

In the preparation of this paper the underlying thought has been to treat the subject in an impersonal manner. Illustrations and specific reference have been made to our practices only when this has appeared to be the simplest and most direct method of approach.

The Unwisdom of Overloading

We believe this question is of paramount importance, not only to the automotive industry but to all who are contemplating bus operation in any form. Our policy is predicated on a seat for every passenger. At the inception of our business this was our slogan. We have never departed from it and we never expect to do so. We are convinced that this policy has been, perhaps more than anything else, a factor in the building up of our enterprise.

It is, of course, possible to carry a certain percentage of standees in a vehicle, the spring-suspension of which has been correctly designed to carry properly a seated load. In our judgment, however, this figure should not exceed 30 per cent. But even this is unsatisfactory, for once standees are permitted, their limitation is most difficult.

Obviously, the problems requiring solution from the standpoint of spring-suspension are much less numerous with vehicles operating on rails than is the case with rubber-tired equipment running over roads. With the former, overloading has no immediate serious consequences—at least from the standpoint of the rolling stock. The spring-suspension with a bus must of necessity be a compromise between minimum and maximum loads. If the range is too wide, bad riding conditions must obtain during by far the greater percentage of the total time, for the packed loads will, generally speaking, occur only during the rush periods. This means that 90 per cent of the time there will be a state of discomfort. This will have an extremely bad effect on both the vehicle and its occupants. Another vital point to consider is that a bus is not kept in a comparatively straight and rigid course by steel rails. The advantageous flexibility of a bus in steering its course at will has its disadvantages if standees are permitted, for the shifting of the weight of the standees when the bus swerves tends to make it unsafe, throwing the passengers about inside the vehicle and rendering the operator liable to heavy damage and accident suits.

We are unqualifiedly behind any movement that will aid the bus to come into and remain in the field that is peculiarly its own. We are positive that the short road is the seated load and if builders will bear this in mind from the standpoint of design and warranty, the automotive industry will assuredly find ample repayment.

We earnestly hope that the automotive industry will read the writing that is so plain to see and that it will profit by what has occurred with the street railways, in regard to the matter of overloading. For it must be remembered that the bus has its limitations and that it is not the cure-all for every ill that transportation is heir to.

THE MATTER OF FARES

Strictly speaking, there is no actual relationship between the design of a bus and the fares charged to passengers. Obviously, however, the better the design, the lower will be the operating cost. Naturally, this will make for lower fares. We believe that in the present state of the art no real success can be attained with less than a 10-cent fare. We are, of course, assuming operation based on seated loads and ample service during both the light and the heavy hours. But with this character of service and with equipment that is properly designed and maintained, the people are very willing to pay a 10-cent fare. There is ample evidence of this in New York City, Detroit, Chicago, Toronto and other cities.

The necessity for a 10-cent fare does not rest with the bus alone. Many electric railways need a 10-cent fare in order to be put on a paying basis. The last available tabulation shows that 140 electric railways in the United States are receiving a 10-cent fare, and that over 95 per cent of the electric railways in the cities of the United States have received varying increases in fare during the last few years. Some cities have a first fare of only 6 or 7 cents, but to this must be added a charge for transfers. Many cities have been placed on the zone system that works out in some cases as high as 3½ cents per mile. Even with an increased fare, the last available figures show that about 10 per cent of the electric railways in the United States are in the hands of receivers.

It is not the purpose of this paper to enter into a lengthy discussion of operating costs, for unless this matter is treated in considerable detail, accurate deductions are almost impossible. Obviously, a correct comparison of operating expenditures can be made only on the assumption that similar detail classifications are employed in conjunction with a similar accounting system. Here the difficulties begin, for as yet few companies operating buses use the same accounting methods.

No doubt there are many who, while not desirous of making a minute survey of details of operating costs, would be interested in knowing something about this rather complicated matter other than mere expressions of opinion. For this reason there is shown in Table 1 not the customary detail cost statement, but what might be described as an income analysis. Actually it represents a distribution of the dime received from each of those who rode on our buses during the year 1921.

TABLE 1—DISTRIBUTION OF EACH FARE RECEIVED

	Cents
Total Operating Expenses	6.50
Total Taxes	1.16
Reserved for Injury and Damage Claims	0.17
Reserved for Depreciation	0.29
Interest on Capital Investment	0.39
Net Income	1.49
Total	10.00

From these figures it is abundantly clear that we should have made a very bad showing with a fare of less than 10 cents. Here is emphasized very clearly the fact that the success or failure from the standpoint of an undertaking such as our own depends absolutely on the addition or subtraction of what at first sight appear to be insignificant amounts. To emphasize this point, during 1921 we carried a total of 52,216,946 passengers, so the net income from this source at 1.49 cents per passenger works out at $778,032.50. To permit of a comparison being made between the conditions confronting us and those faced by others, it should be noted that we operate a total of 25 miles of one-way route, that our longest run is 10.2 miles and our average haul 5.0 miles.

THE BUS AND ITS SERVICE REQUIREMENTS

Before discussing the bus from a design standpoint, something may be gained by outlining the character of service that must be expected, for it is here that the average engineer underestimates the difficulties to be encountered. First, let us consider the cumulative result of a year's performance of the physical limitations that are primarily responsible for wear-and-tear. For the sake of argument it may be assumed that these data are applicable to any bus operated by any public utility. The figures are presented in Table 2.

Assuming the same general plan of upkeep as employed by the Fifth Avenue Coach Co., each bus would be thoroughly inspected after every 2000 miles of operation and rebuilt and repainted yearly. A vehicle would be expected to require no incidental repairs between inspectional periods and no major repairs between either inspections or yearly overhauls. The inspectional periods would occur approximately every 14 days. The maximum inspectional allowance is 8 hr. The allowance for yearly overhaul is 7 days. Roughly, it may be said that under these conditions, each bus is scheduled for service 358 days out of 365.

TABLE 2—DATA ON BUS OPERATION IN NEW YORK CITY

Yearly Mileage	30,000 to 60,000
Stops and Starts	180,000 to 360,000
Change-Speed Applications	360,000 to 720,000
Clutch Applications	360,000 to 720,000
Different Drivers	1,095 to 2,190
Brake Applications	200,000 to 400,000

The statistics quoted as to mileage, stops and starts, and the like, speak for themselves. Those who have never had control of a public utility operating buses cannot possibly picture the sum total of the abuse the average bus must suffer. More than anything else, frequent changes in drivers result in increased service difficulties. It may be safely said that if one could with a bus have the same driver daily, at least 50 per cent of the service troubles would disappear. This, however, is quite impracticable, since the loss of earnings would be many times the decreased service cost. Even with an operation of moderate size, the bus must of necessity lose its identity. It becomes merely a transportation unit. There must be changes of drivers daily, many of whom will feel scarcely any pride of ownership. All they are concerned with is being on schedule time. This means that the bus will be subject to extraordinary abuse. The mechanisms of the bus must be capable of withstanding treatment of the most brutal nature; otherwise constant failures will occur.

Before one can proceed very far from a design standpoint, there must be some fairly clear conception of the vehicle life that is to be expected. In this connection it is necessary to lay stress on the fact that motorbus design is still in its initial stages. Five to 7 years is about the maximum life of the most modern type. It is not a matter of wear-and-tear, for a vehicle may be so well cared for that there is no limit to its life. Obsolescence is the real issue. The ideal conception is to carry out the design so that the various units, which when assembled comprise the complete structure, have as nearly as possible an equal life.

CONTROLLING DESIGN FACTORS

In its broadest sense we believe the controlling design factors from the standpoint of the motorbus, in the order of their importance, are

(1) Safety
(2) Comfort and convenience of the public
(3) Minimum operating cost

Safety easily heads the list and a very large proportion of the engineering development work must be concentrated under this heading. It is generally agreed that a truck carrying freight should be in all respects safe, and that every reasonable precaution should be taken to render automobiles transporting from 1 to 7 passengers safe; so how much more important is it that a vehicle carrying 50 or more passengers should be free from every sort of hazard! It must be remembered that much of the mileage of the bus is through congested thoroughfares. This is not the case with the average automobile or truck. Again, the average individual makes some effort to get out of the way of a truck or automobile, but the bus, with its acknowledged flexibility, is supposed to move out of the paths of both vehicles and pedestrians.

The design of a motorbus from a safety standpoint includes certain basic features which must be incorporated in the general constructional plan. There are also other detail features which must be included. The latter are dictated by human considerations. Reference is now being made to providing the driver with reasonable comfort and convenience so that no undue hardship will be inflicted upon him as a result of the performance of his duties. First, let us consider the former. These are

(1) Low center of gravity
(2) Wide frame, track and spring centers and general dimensions
(3) Effective brakes
(4) Short turning-radius

Low Center of Gravity

Beyond doubt, the future bus will be low hung. The inherent danger in connection with any other form of construction is the possibility of overturning. Under conditions of proper operation, the hazard may be nonexistent, but we have always before us the possibility of human failure. Actually the danger is much more real than apparent. The controlling element governing overturning is centrifugal force. Vehicles seldom if ever overturn as a result of high speed and sudden impacts or brake applications. Overturns are almost invariably due to a combination of speed and turning-radius. The only reliable guarantee against this class of accident is a low center of gravity.

In many cities there are overhead wires and various other obstructions. The low bus is often a necessity to pass under such obstructions. Certainly, the lower the vehicle, the less the hazard. These remarks apply particularly to double-deck vehicles. With the single-deck vehicle, the higher speed is a factor that must be fully taken into account. Entirely apart from the matter of safety, a low-hung vehicle has a more graceful appearance. There is less time lost in boarding and alighting, there are fewer boarding and alighting accidents, and the schedule speed can be faster. Lastly, assuming proper design, a low center of gravity results in improved riding properties.

We have found that a safe and practical height of the frame from the ground for a single-deck bus is 25 in. and for double deck bus, 18 in. The center of gravity of our type-L double-deck vehicles, with a full complement of passengers on both decks is 52 in. from the ground. With our type-J single-deck bus, this dimension is 38 in. It is interesting to note that when rounding corners, even at a high rate of speed, skidding will occur due to cen-

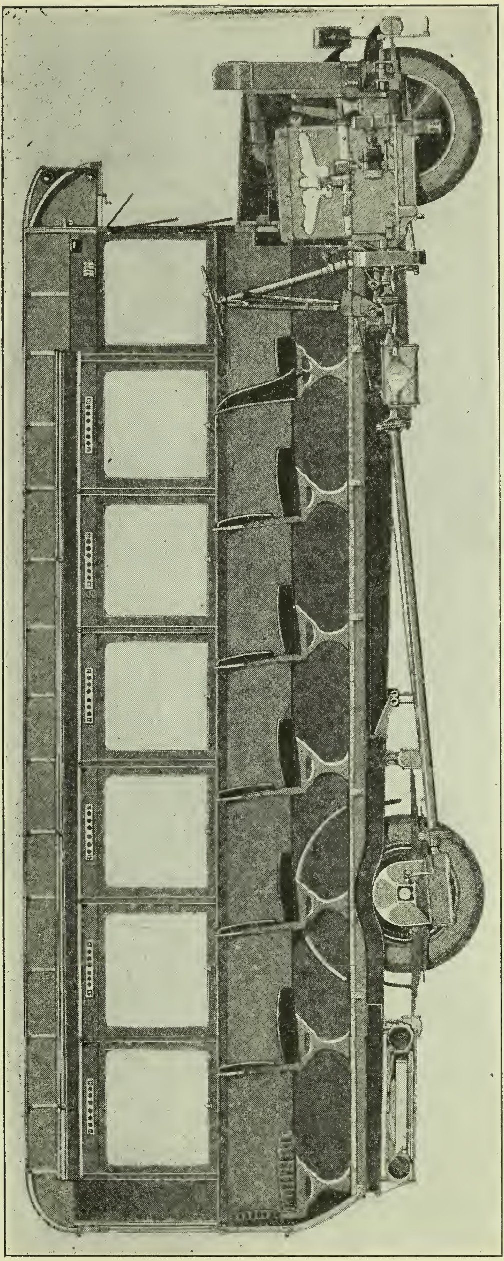


Fig. 1—Sectional View of the Type-J Bus

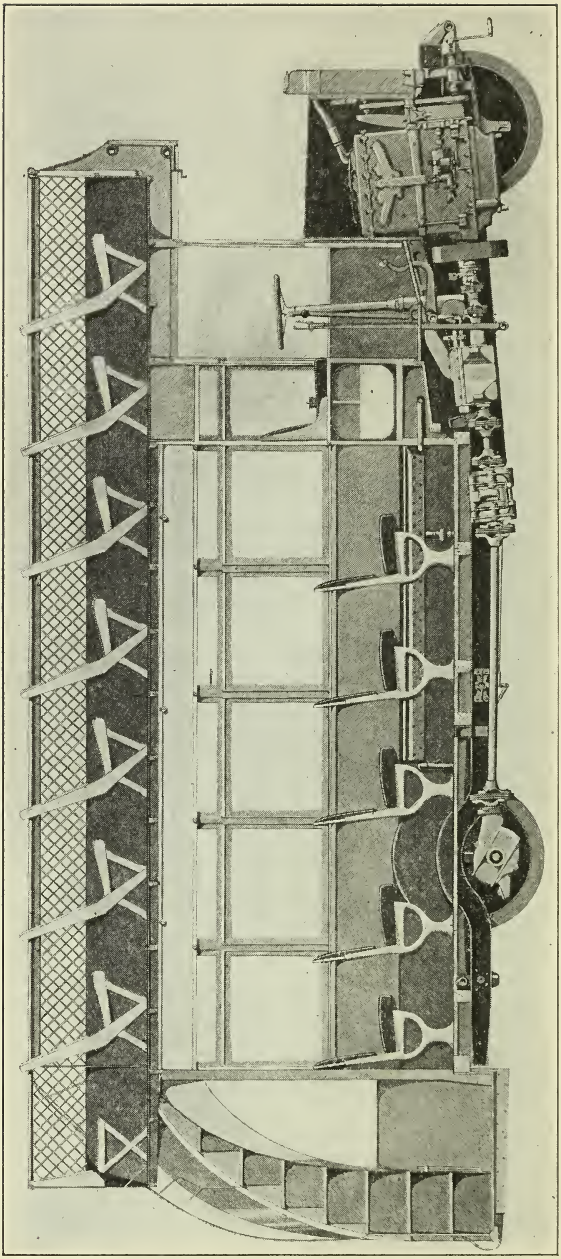


Fig. 2—Sectional View of the Type-L Bus

trifugal force and overturning is scarcely possible. Furthermore, rolling or sidesway is practically eliminated. The sectional views of our J and L-type buses reproduced in Figs. 1 and 2 indicate clearly how this condition has been reached. With type L it will be seen that the frame and rear-axle construction is somewhat unconventional. The rear axle is of the internal-gear type. The spiral bevel-gear and differential assembly is in unit form and can be entirely assembled and adjusted on the bench. The carrying member is a heat-treated forged job.

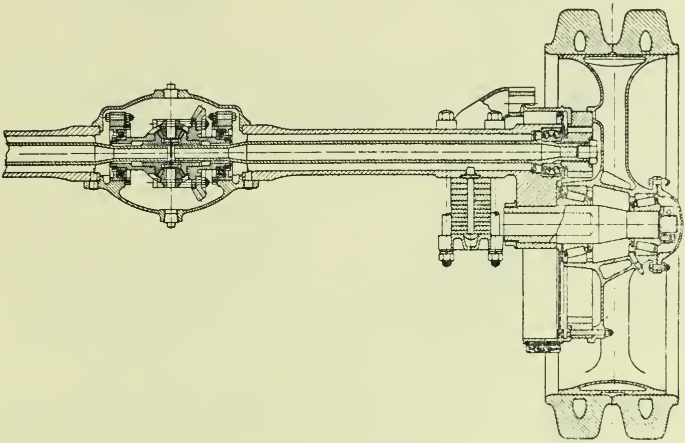


FIG. 3—SECTIONAL DRAWING OF THE TYPE-L AXLE

From the sectional drawing shown in Fig. 3 the general construction of the type-L axle will be clear. It will be seen that the ends of the carrying member are cranked, the wheel spindles being above the drive-shaft center-line. It is in this manner that the low-level feature has been accomplished. The photograph showing the carrying member and driving-gear assembly which is reproduced in Fig. 4 at once emphasizes the general simplicity and accessibility of construction. Due to the fact that the drive-shaft pinions are in the vertical plane, a special form of tooth has been developed for the internal gear to provide adequate clearance and at the same time permit of maximum silence even after a certain amount of wear has occurred.

We do not employ this special form of axle construction for the type-J bus. This class of vehicle will have a much wider use; therefore, the matter of road clearances must be taken into account. In many cases single-deck vehicles will be operated over very bad roads. The double-deck vehicle is essentially a city job where the streets are, generally speaking, in fair condition. Again with the single-deck vehicle, the floor-level requirements are not so exacting. There is no top deck to take care of, and the entrance can therefore be located at the front end of the bus; but with the double-deck vehicle, conventional practice is to have the passengers enter at the rear, so in passing to the interior they are obliged to cross the rear axle which must be of special design to have the floor level within easy stepping distance of the ground. In the case of the single-deck bus it is not desirable to have a step 18 in. high. Therefore, the best plan appears to be to employ an orthodox rear-axle design. Even assuming the use of our type-L rear axle, it would not be practical to produce a stepless vehicle. The appearance would be completely spoiled and, as explained above, the ground clearance would be cut to a point where the vehicle would be unsuitable for use in many localities. Of course, a stepless single-deck vehicle can be produced, but its practical value for general utility purposes is debatable.

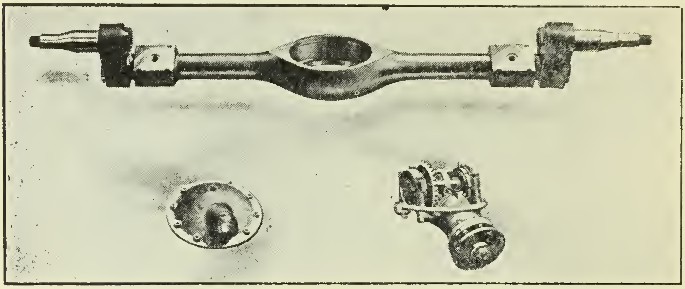


FIG. 4—THE TYPE-L REAR AXLE

Among the constructional difficulties in connection with the production of low-level equipment, one of the problems is to obtain a flat floor. There is a natural tendency for the components to project above the frame and therefore through the floor. To avoid this, special design is required. The effect of a flat floor is very pleasing to the eye. Its structural strength is greater. It is less costly to keep in repair and there is less possibility of accidents due to the passengers' feet coming into contact with the

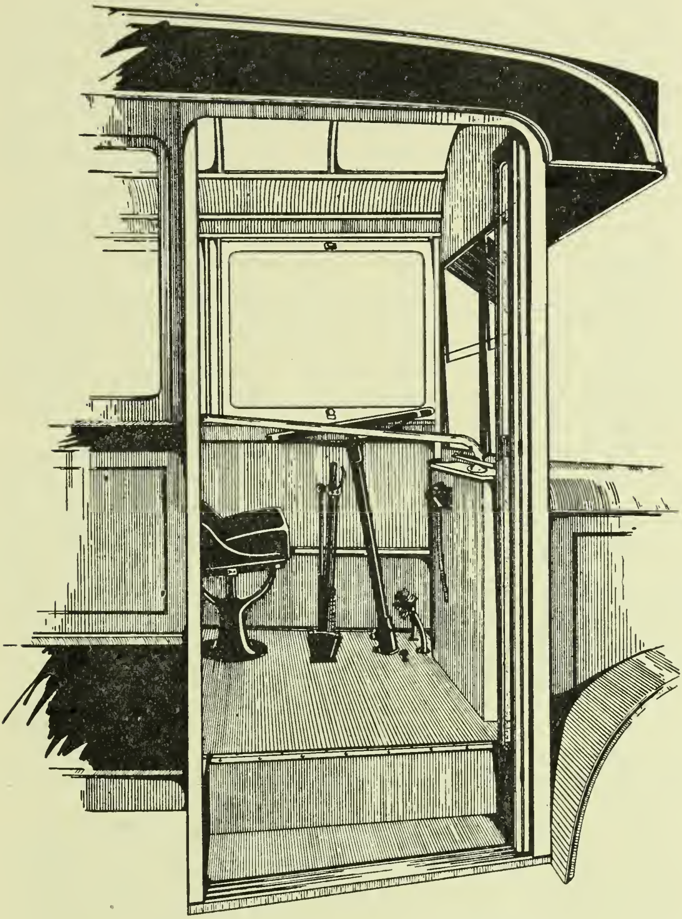


Fig. 5—View through the Door of the Type-J Bus

obstructions during the boarding and alighting processes. The view through the door of our type-J vehicle, Fig. 5, brings out this point to advantage.

WIDE FRAME, TRACK AND SPRING CENTERS

These features are necessary to provide for adequate vehicular stability and, in conjunction with a low center of gravity, make for maximum safety. The necessity of providing proper stability applies equally to single and double-deck vehicles. It may be said that the added risk due to the top-deck load with the latter is more than equalled by the faster speed of the single-deck unit.

Apart from the matter of safety, a wide frame is necessary in connection with the body construction. Obviously it is desirable to support the body as far out as possible, for in all cases the seating arrangement is such that the passengers are grouped about the outer edges. Then, the wide frame admits of the lightest possible form of body under-frame. The wide frame also is a factor from the standpoint of the passengers' comfort. This point will be referred to later.

We believe that the overall length of a motorbus for city service should not exceed 26 ft.; the total width, 7 ft. 6 in.; and the over-all height for single-deck vehicle, 9 ft. With the double-deck bus, the last-named dimension should be such that a person standing on the top deck can clear a 14-ft. structure. With these dimensions we have found it possible to accommodate comfortably 51 seated passengers with our double-deck, and from 25 to 29 with our single-deck vehicle. Whether this practice is economically correct for all localities, we cannot say. We have, however, up to the present found that this arrangement works out very well both in our own service and in the service of those who have purchased our equipment.

Next, there is the question of important dimensions other than those over-all, such as the wheelbase which naturally affects the axle load distribution, the turning-radius and the general comfort and balance of the vehicle. For the class of vehicle now under discussion, we believe that this dimension should not be less than 168 nor more than 180 in.

The front track should be ample in width and not less than 67 in., for to turn a 'bus within the intersection of the average city street, it is necessary to move the front wheels through an angle of not less than 35 deg. This determines the distance between the front-axle pivots and the springs. The spacing of the front springs should not be less than 36 in., since they are responsible to a large extent for the stabilization of the vehicle when turning a corner.

Regarding the rear track, we believe that the outer edge of the tires should closely correspond to the extreme over-all width of the body and that the rear springs should be as close to the tires as is practicable. For buses as above described, the rear track should not be less than 72 in. This will bring the distance between the springs to approximately 52 in. Having decided the approximate distance between the vehicle springs, it naturally follows that the best design is to arrange the frame dimensions so that they connect with the springs in the closest and most practical manner. Our practice in regard to these matters may be readily followed from the diagrammatic sketch of the type-J chassis as shown in Fig. 6.

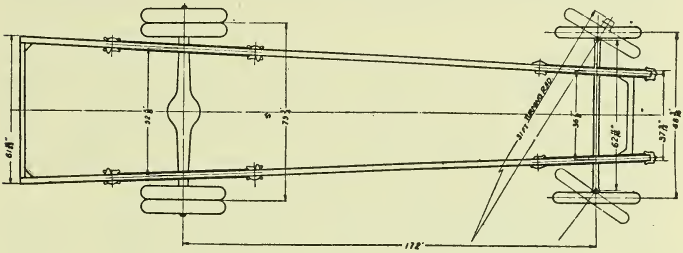


FIG. 6—CHASSIS DIMENSIONS OF THE 29-PASSENGER SINGLE-DECK BUS

EFFECTIVE BRAKES

Perhaps the most difficult problem that engineers must face is the brake question. Even now it has not as yet been solved entirely satisfactorily, at least insofar as our knowledge goes. With the bus, the number of applications is in excess of that of the average truck or automobile, and the brakes of a bus must be sufficiently powerful to lock the wheels at any moment. Yet the effort required for average application must not be such that a driver may become exhausted as a result of the work imposed upon him.

Particular attention must be paid to the location of the hand-brake lever. It should be placed so that it can be grasped firmly without moving the body out of the

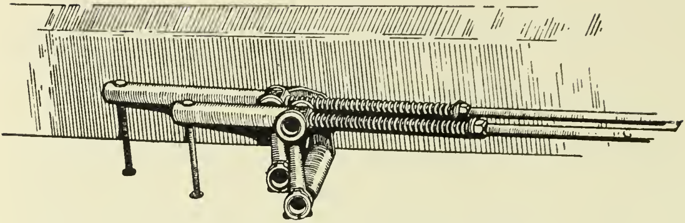


FIG. 7—OUTSIDE BRAKE ADJUSTMENT

normal seated position. We believe the best practice is to have the lever arranged for a push and not a pull-on. Time can thus be saved, and a fraction of a second is often the determining factor from an accident-prevention standpoint.

The brakes of a bus must be free from undue noises such as squeals or rattles. This means, among other matters, the use of special brake-drum material. The conventional soft pressed steel is practically useless. The best plan is to employ treated steel forgings or, failing in this, steel castings with a high carbon-content.

The friction surfaces must have long life, and the adjustment be such that no tools or special skill is necessary. We attach considerable importance to the matter of foolproof adjustment. The J system as illustrated in Fig. 7 shows our method. It will be seen that there are two vise-like levers. The outside controls the hand, the inside the foot brake. One turn is usually sufficient. If by any chance the levers are not returned to the vertical, they will automatically reach this position by force of gravity.

The braking action must not be too abrupt. It must be positive yet not sudden and violent, for such a condition is exceedingly severe on the driving members, tires and body. It is also a frequent source of accidents from which serious claims may result. Brakes must be sufficiently good, yet not too good. Excessively efficient brakes have a most marked influence on tire wear. It may be said that tire wear is almost directly proportional to the effectiveness of the brakes.

In bus operation it is desirable from every point of view to cover the route as quickly as safety will permit. In this manner the maximum number of passengers can be carried daily. With a fixed maximum-speed, this means fast deceleration and acceleration. Expressed in another way, the problem is to move from a stop in one location to a stop in another in the least time. In our own service this must be done without exceeding a speed of 15 m.p.h., or accelerating or decelerating faster than 2 m.p.h. per sec. per sec. A more rapid rate of deceleration is, of course, available for emergencies, but it will be uncomfortable and unsafe, especially for standees.

The acceleration and deceleration graph as reproduced in Fig. 8 shows how closely the present type of equipment approaches this conception. But in connection with a study of this graph, the following points should be borne in mind:

(1) The bus was fully laden, sand bags being used instead of passengers

(2) The test was carried out on upper Broadway where traffic is not heavy

(3) Normal service conditions were followed, no attempt being made to obtain maximum acceleration or deceleration

(4) A bus was selected at random for the test

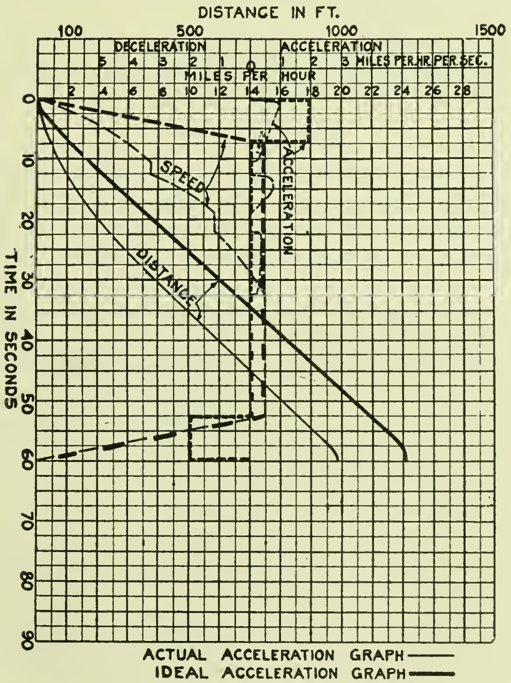


Fig. 8—Curves Showing the Actual and Ideal Acceleration

SHORT TURNING-RADIUS

One of the great advantages of a bus over any other form of transportation unit is its flexibility. A bus can be switched around at any point, and it is highly desirable that it should be able to make a complete turn in the average thoroughfare without backing, for the latter practice if followed in congested areas merely adds to both confusion and congestion. There is also a marked possibility of an increased number of accidents.

A short-turning-radius is dependent on the interference of the tires with the drag-link, front springs or frame, when the wheels are turned at the maximum angle. The controlling elements are wheel-spring tracks and wheelbase. As the radius of the steering angle equals the wheelbase divided by the sine of the front-wheel lock, it can be seen that a wheelbase of reasonable length is important to secure a short turning-radius.

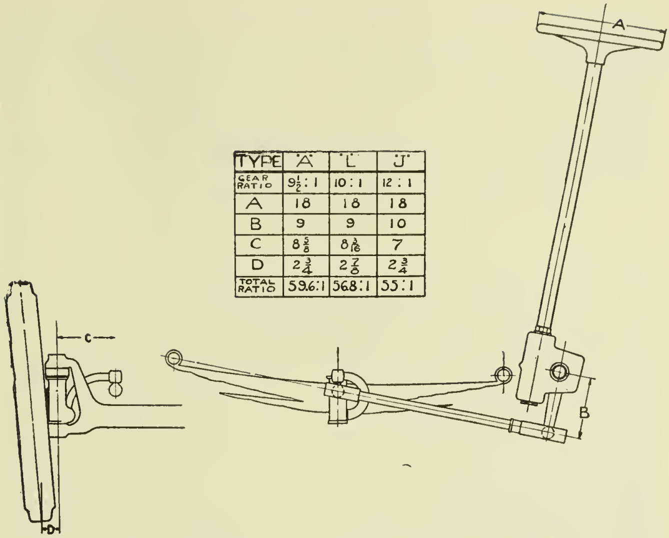
From the viewpoint of safety, the design features dictated by human considerations are

(1) Easy steering
(2) Clear vision for driver
(3) Comfort and convenience for driver

EASY STEERING

The steering of a bus should be at least as easy as that of the average automobile. To operate a stiff steering-gear is a hardship that certainly should not be inflicted upon the driver of a public-service vehicle. A driver's energy and effort must be concentrated on his regular duties, and if he becomes fatigued through the expenditure of unnecessary effort, faulty operation is bound to result. This means possible accidents. Tests have convinced us that the actual physical labor imposed on the driver of a bus in connection with the manipulation of a steering-wheel represents by far the greater proportion of the sum total of his work.

Ease of steering is controlled by the total ratios between the hand and road wheels. Naturally frictional losses in the steering-gear box and steering-knuckles are of importance. Minimum losses in these respects are dependent upon the use of properly lubricated antifriction bearings. Another very important matter is that the steering-knuckle pins should lie in the vertical plane; otherwise there will always be a tendency to lift the front



TYPE	"A"	"L"	"J"
GEAR RATIO	$9\frac{1}{2}:1$	10 : 1	12 : 1
A	18	18	18
B	9	9	10
C	$8\frac{5}{8}$	$8\frac{3}{16}$	7
D	$2\frac{3}{4}$	$2\frac{7}{8}$	$2\frac{3}{4}$
TOTAL RATIO	59.6:1	56.8:1	55:1

Fig. 9—Diagram of the Steering Leverages

end of the bus when turning the steering-wheel. An angle in either the longitudinal or transverse plane will cause lifting at the expense of effort on the part of the driver.

It is highly desirable that there should be an absence of shocks at the steering-wheel. This is largely controlled by the total ratio, but also by the distance between the point of contact of the wheel and the road and the intersection of the knuckle center-line and the road. Every effort should be made to keep this distance small. With the J type the length of the lever arm is about 2¾ in.; and an increase of only 1 in. would decrease the total ratio some 36 per cent. This is the only point in the steering linkage where a change increasing the total reduction does not result in increased steering-wheel travel for a given lock. A short drag-link or the incorrect alignment of the drag-link with the front springs will also result in shocks at the steering-wheel when passing over rough roads.

Minimum steering-wheel travel is important as it makes a change of hand position unnecessary for ordinary driving. It also decreases the apparent back-lash, which is present in all steering mechanisms. The steering-wheel travel is roughly inversely proportional to the total ratio, which is kept as low as possible for this reason.

Our practice so far as the important dimensions referred to above are concerned may readily be followed from an examination of the diagram of steering leverages as illustrated in Fig. 9.

CLEAR VISION FOR DRIVER

This very important feature can be accomplished only as a result of joint chassis and body design. The driver should be located close to the left-hand side. This permits him to observe and also to signal his intentions to oncoming traffic. There should be absolutely nothing obstructing his view. He should face clear glass. It should also be mentioned that with single-deck vehicles the placing of the driver well over on the left-hand side provides for the very necessary boarding and alighting space for passengers and adequate room for operation of the door.

Briefly, a driver's vision should be such that when seated, even back of a closed windshield, he will have nothing on which he can readily concentrate, no vertical posts or obstructions of any kind. He should just naturally sense that he is in the open. The illustrations of the front end of our type-J bus reproduced in Figs. 10 and 11 bring out this point with marked clearness.

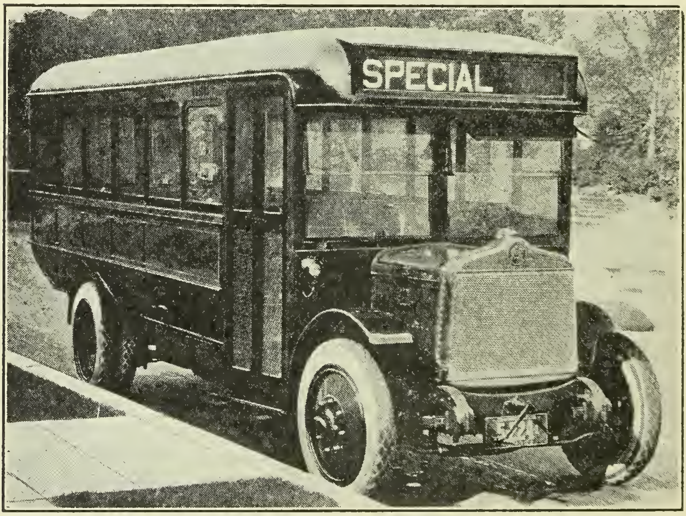


FIG. 10—A TYPE-J 25-PASSENGER SINGLE-DECK BUS

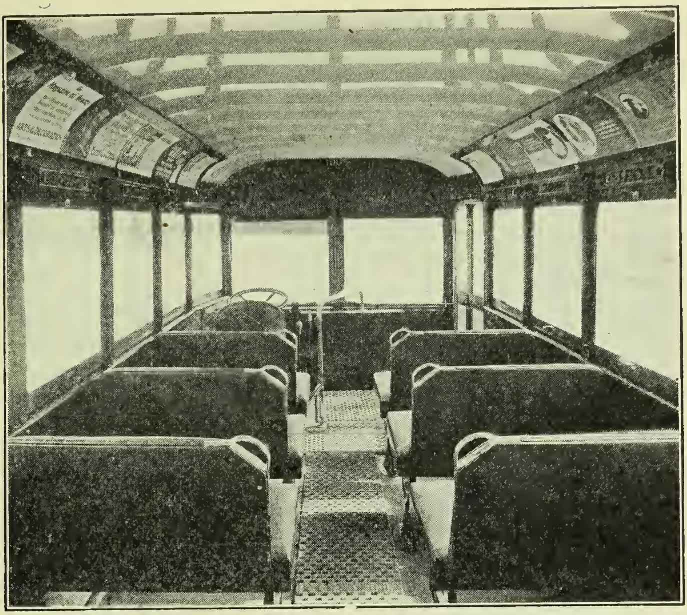


Fig. 11—View from the Rear End of a Type-J Bus

Comfort and Convenience for Driver

This is largely a question of seat formation in conjunction with the correct positions for brake, change-speed levers, pedals, accelerator, etc. Obviously, it is not a practicable matter to give the driver of a bus as much room as with a touring car; therefore, much care and thought must be paid to the placing of pedals and levers. The conventional cowl as used in automobile practice is almost out of the question, for anything that tends to increase the over-all length of the vehicle is distinctly undesirable, particularly if such increases add nothing to the passengers' seat or pay-load space.

The driver should be comfortably seated at all times. He should be able to reach his change-speed or brake levers without body movement. He should have ample leg-room and not be obliged to cramp his limbs when his feet are either on or off the pedals. To some extent this point is brought out in Fig. 5. The value of the flat floor from the standpoints of both passengers and driver, is apparent; also the side control without which there is of necessity a considerable loss of most valuable space.

COMFORT AND CONVENIENCE OF THE PUBLIC

The American public is automotively inclined and the percentage of those owning cars is so large that when riding in any self-propelled vehicle, there is a natural tendency to compare its behavior with that of an automobile. In designing a bus this factor must under no circumstances be lost sight of. The success of any public utility depends on the good will of the public. It has been correctly stated that the permanence of any business depends upon the good will of those it serves and that no business can achieve permanent success that does not give in exchange for its earnings at least an even measure of helpful service. This applies especially to public utilities, and the truth has been abundantly proved in connection with the operation of our enterprise.

From the viewpoint of design, it is essential that consideration be paid to the attitude of the public as a whole. It is not enough to consider only the attitude of the actual riders; regarding the matter of comfort from these somewhat different angles, it is necessary that attention be given to

(1) Riding ability
(2) Reliability
(3) Silence of operation
(4) Smoothness of starting and stopping

RIDING ABILITY

Broadly, this is a matter of proper spring-design. There are, however, other important influences; the wide frame, track and spring-centers bear materially upon this question, for the nearer the wheels are to the outer edge of the body, the less will be the movement to which passengers must be subjected when obstacles are passed over. Again, with the wider track, many of the ruts and depressions created by vehicles of narrower gage, will be passed by. Incidentally, this is quite an important matter from the standpoint of road wear. The wide track also diminishes the wheel-pocket projection inside the body. The modern tendency is to employ cross seats and with the narrow-gage vehicle the wheel pockets are a source of much discomfort to those seated upon the inside immediately over them. A rigid frame correct axle-load distribution and minimum overhang are all factors that make for better riding performance.

Apart from the points briefly touched upon above, the

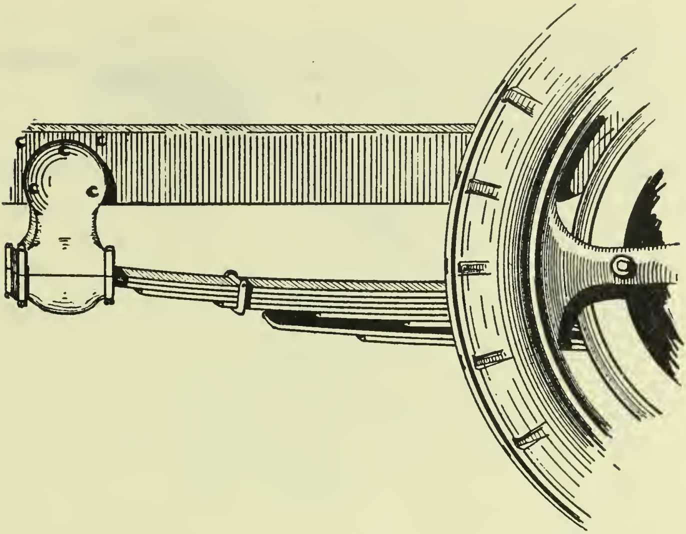


FIG. 12—THE TYPE-J PROGRESSIVE SPRING ARRANGEMENT

controlling factor from the standpoint of riding ability is, of course, the design of the suspension itself. Obviously, the difficulty is to obtain good riding under all conditions of load. Spring design is always a compromise; a spring must be able to withstand maximum load, yet vehicles are expected to ride reasonably well when light. As a matter of fact, they seldom, if ever, do so. In general, more damage is done to vehicles when running light than heavy because the riding properties under these circumstances are at their worst and the speed too often is high. Under conditions of heavy load, springs function best, and at the same time there is less likelihood of excessive speed.

We believe that the answer will be found largely in the employment of what we term the progressive spring as illustrated in Fig. 12. It will be seen that the spring is split into two parts. The top half takes the weight of vehicle, body and a certain proportion of load. The bottom part or helper comes into action progressively. The top part must make a rolling contact with the bottom. One of the great advantages of this system is the fact that, for no additional cost or weight, a marked improvement in performance is possible. The theory behind our choice of the progressive spring and the advantages that may be derived from its employment can readily be seen

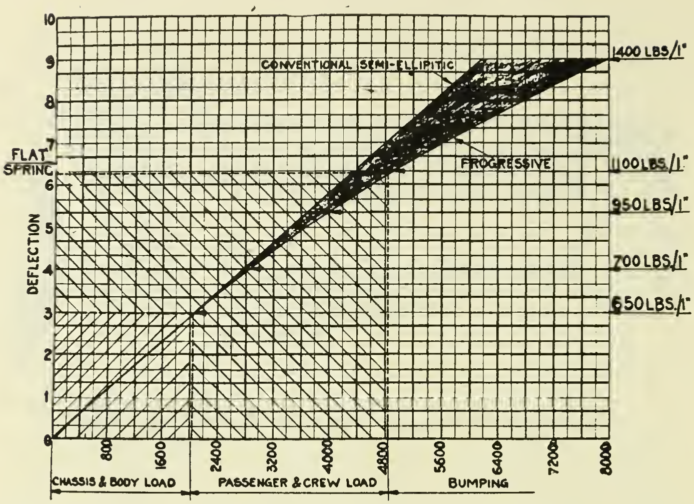


Fig. 13—Curve Showing the Comparative Deflections of the Progressive and Conventional Semi-Elliptic Rear Springs

from an examination of the rear-spring deflection curve for both the progressive and the conventional semi-elliptic designs reproduced in Fig. 13. No doubt it will be appreciated that to secure comfortable riding with a small number of passengers, it is necessary to have a spring of not over 670-lb. per in. deflection. But a spring having these characteristics is not a practical arrangement, for the result would be too great a difference in body and step height between the minimum and maximum number of passengers. This point is clearly shown in the graph where the proportion of the 51-passenger load equals 2800 lb. per rear spring, from which the comparative figures given in Table 3 are deduced.

TABLE 3—DEFLECTION IN INCHES FOR PASSENGER LOAD

	Conventional Semi-Elliptic Spring	Progressive Spring
Full Passenger-Load	4¼	3¼
Maximum Bumping-Load	8½	6¼

The deflection curve of a simple semi-elliptic spring is a straight line showing a constant load per inch. But as the progressive element comes into play gradually, a curve is apparent. The departure from a straight line

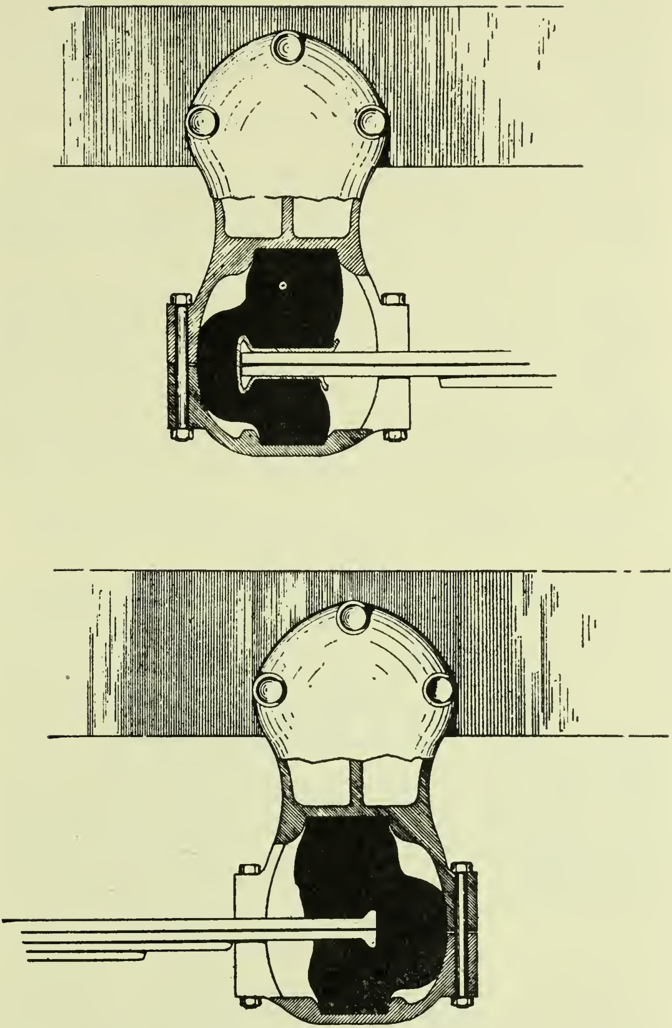


FIG. 14—RUBBER SPRING-SHACKLES USED ON THE TYPE-J SPRING SUSPENSION

which is shown shaded represents the load carried by the progressive element which can be designed to come into action at any desired point. It has been found most satisfactory to design this spring so that the stiffened action begins very gradually, that is to say, after a limited number of passengers have been taken on. Obviously, as the progressive element comes into action, there is a gain in the stability of the vehicle.

From the graph above referred to it is exceedingly interesting to note the change in rate of progression as a result of a variation in passenger load. The figures based on increments of 10 passengers given in Table 4 bring this point out in a striking manner.

For our single-deck equipment we have standardized the Mack type of rubber shock-insulator which is illustrated in Fig. 14. This is by special arrangement with the Mack company. We are experimenting with this device for our double-deck vehicle but as yet are not prepared to state the results. This arrangement, in conjunction with our progressive system, markedly improves the riding conditions. It also avoids the necessity for lubrication and for replacement of shackles, shackle-pins and bushings, also, no spring-eyes are required. Experience up to the present shows that we may expect a very satisfactory life from rubber blocks.

TABLE 4—CHANGE IN RATE OF PROGRESSION FOR VARIATIONS IN LOAD

No. of Passengers	Load per 1-In. Deflection, lb.	Increased Stiffness, per cent
0	670	0.0
10	780	16.4
20	810	20.9
30	850	26.9
40	900	34.4
50	1,080	61.3

SILENCE OF OPERATION

It is a problem to produce a silent vehicle. It is doubly a problem to retain this state throughout the life of the vehicle. Silence necessitates freedom from engine vibration, quiet transmission gears, evenly stepped gears, a quiet rear end, and generally the elimination of all rattles and squeaks from both body and chassis. To attain this, every detail of design must receive the most minute care. Silent operation is necessary in crowded thoroughfares, and certainly the people demand this condition in the residential areas, particularly at night when the streets are comparatively empty and noises become automatically emphasized. As a rule, noises are tolerated simply because such things are nearly always with us, but in the quiet of the evening sounds that ordinarily pass unnoticed become startlingly evident. In connection with the general question of noise it is interesting to consider for a moment conditions on Fifth Avenue in the rush period during which we operate 180 buses per hr. in each direction. If this vehicular volume were not reasonably quiet, we should soon be ordered off the streets as a public nuisance and a menace to health.

From the standpoint of silence, our greatest difficulty has been and still is the matter of transmission gears. We use the four-speed gear and the three-speed chain transmission, shown in Figs. 15 and 16 respectively, depending upon the class of service and general operating conditions. It will be seen from an inspection of Figs. 17 and 18 that the shift-rods, their bearings and the lock mechanism are of substantial proportions.

Illustrations of the speed curves are presented in Figs.

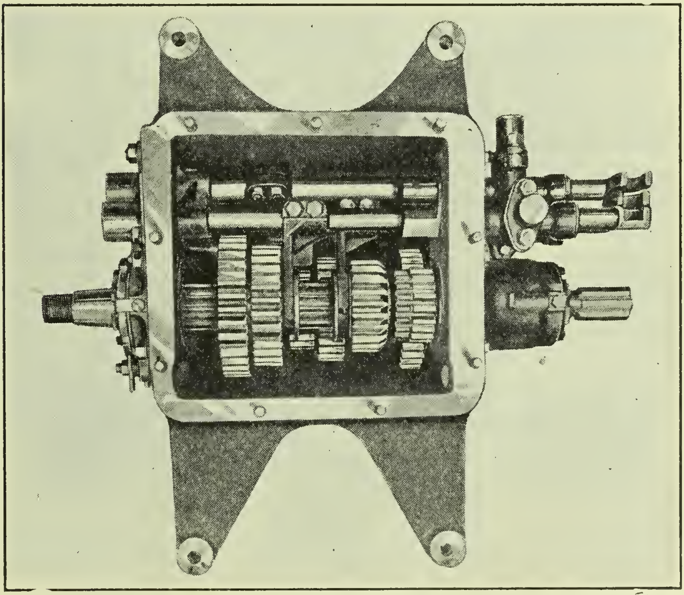


Fig. 15—The Four-Speed Gear Transmission

19 and 20. It is worth while noting that the ratios of the four-speed transmission are almost exactly in geometrical progression. The three-speed transmission is not so satisfactory in this respect but here a compromise is of course necessary. This remark applies to all three-speed jobs. Where grades are severe, four speeds are highly desirable, to cut down ability losses to the minimum. But where roads are practically flat, the advantages of a four-speed transmission are not nearly so marked.

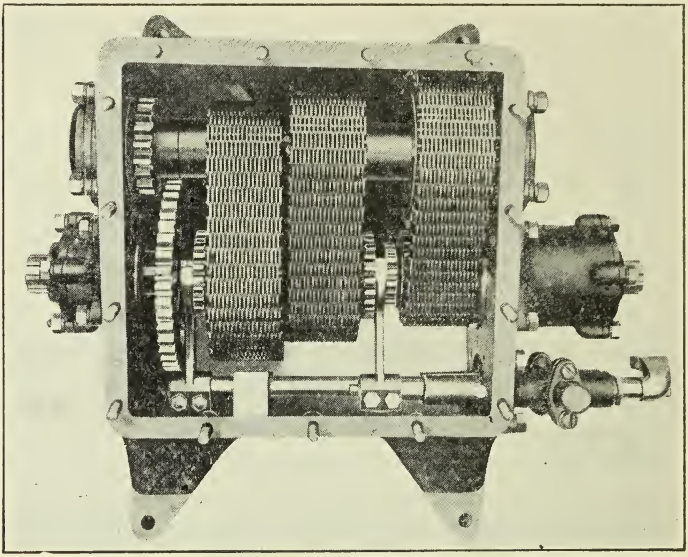


FIG. 16—THE THREE-SPEED CHAIN TRANSMISSION

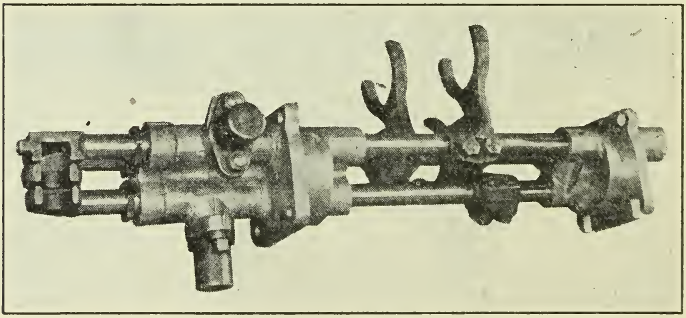


FIG. 17—THE SHIFT-ROD ASSEMBLY OF THE FOUR-SPEED TRANSMISSION

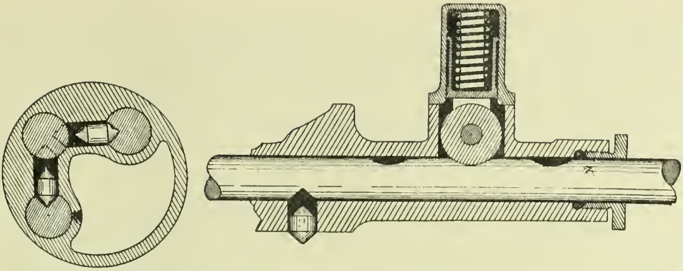


Fig. 18—The Locking Mechanism Used on the Shift Rod of the Four-Speed Gear Transmission

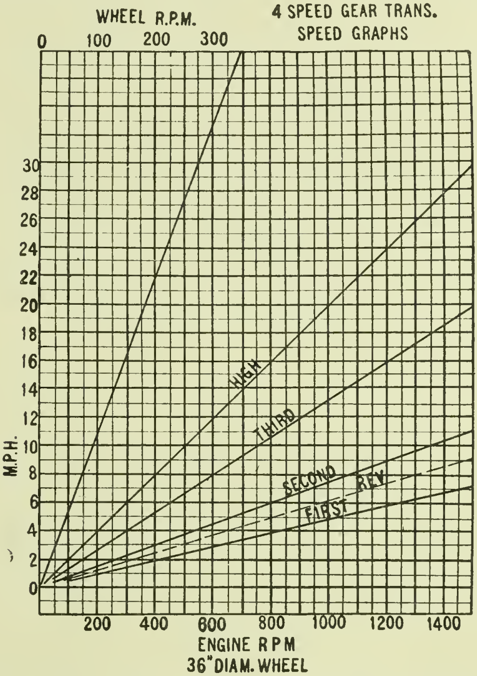


Fig. 19—Speed Curves of the Four-Speed Gear Transmission with a Rear-Axle Ratio of 5.4 to 1

The silent-chain transmission is particularly useful for city service where there are frequent stops and starts, and where the percentage of direct-gear operation is relatively small. Substantially it is similar to a constant-mesh gear transmission but chains are used in place of gears. The shift is extremely short and very easy to effect. Such transmissions remain quiet throughout their useful life, and from our observation one can expect at least a year's service from the chains, which are cheaper to replace than gears. Chain transmissions are standard practice for London bus service.

Reliability

The word "reliability" with a bus attains an entirely new meaning. The entire design must be predicated on ability to give uninterrupted service between clearly defined periods, preferably based on mileage. The ability of a bus to fulfill this requirement with particular reference to the duration of period will at once determine the utility of the design. The public will not long tolerate an unreliable service. Failures with an automobile cause confusion enough but the number of persons involved as compared with a bus is relatively insignificant.

One point it is especially desired to bring home is that under average conditions, drivers cannot be expected to make any attempt whatever to spare their equipment. All they are concerned with is stopping for passengers, avoiding accidents, and keeping in their places on the road in accordance with their schedule. Everything must be subordinated to these three things, and in cases where vehicles cannot stand up under such conditions, either the required changes must be made to enable them to do so or they should be scrapped, for assuredly they have no place in the operation of a public utility.

Smoothness of Starting and Stopping

Smoothness of starting is primarily a clutch function, but of course the driver is a factor. Correct gear-ratios, a satisfactorily performing engine and proper axle-load distribution are contributing influences. Quick starts and stops are highly dangerous from the viewpoint of possible accidents. Some of the heaviest claims for injuries and damages result in this manner. Apart from injuries to passengers, quick starts and stops do more toward causing damage to the chassis and the bodies than anything else. All driving members are subject to abnormal stresses with the former. With the latter, the fore-and-aft or lateral movement, which of necessity results, causes a loosening up of post joints, panelling, etc., and consequently a very high rate of depreciation.

Of the various features that make for efficient and economical operation, the clutch is perhaps one of the

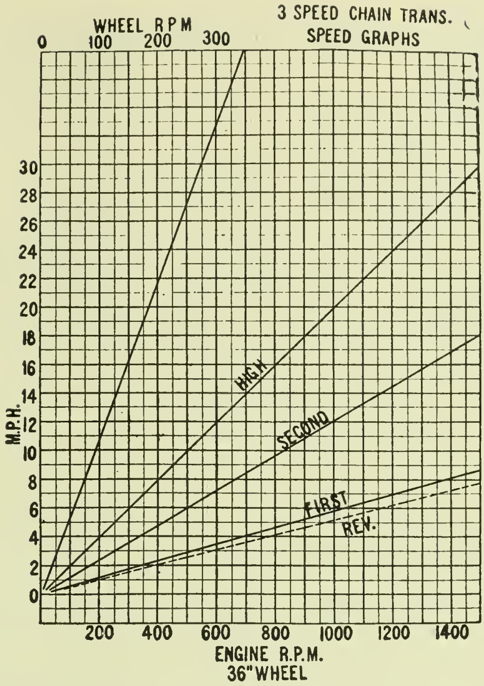


Fig. 20—Speed Curves of the Three-Speed Chain Transmission with a Rear-Axle Ratio of 5.4 to 1

most important. We use exclusively a clutch of the single-disc type. From Fig. 21 it will be seen that there are several unconventional features. Particular attention is drawn to the fact that the spring pressure is evenly distributed over the entire surface of the friction members by 20 small springs, the levers are balanced against centrifugal force and the disc is exceedingly light, thus simplifying the changing of gears. Incidentally, a clutch-stop has been found unnecessary. The removal of the clutch body is an extremely simple operation, as is also the adjustment of the levers. Smoothness of stopping is discussed under the heading of Brakes.

Minimum operating cost demands:

(1) Maximum accessibility

(2) Minimum consumption of labor and material. This of course means excellence of both materials and workmanship

(3) Minimum consumption of fuel
(4) Minimum weight, particularly that which is unsprung
(5) Maximum safe speed. This naturally comprehends rapid acceleration
(6) Maximum tire-mileage

Maximum Accessibility

It is fundamentally necessary that the design of a motorbus be such that inspection and repairs can be carried out quickly and economically. We believe it is imperative that separate unitary construction be followed. For instance, engines, carbureters, all electrical equipment, fans, clutch couplings, transmissions, control levers, axles, wheels and propeller-shafts should all be entities unto themselves, so that the repair of any one of these assemblies will not necessitate the removal of any other.

As a practical illustration, take the orthodox unit powerplant and assume it is necessary to renew the clutch friction linings. The propeller-shaft, transmission and complete control system must first be taken down, possibly even the engine moved forward. In all probability the vehicle must lose a complete day's service. Compare this for a moment with the relatively simple operation where the separate-unit form of construction is employed, such as with our J or L types. Here we need only remove a few bolts from the clutch coupling and housing. The clutch can then be taken out as a complete unit and the linings replaced within a period of 20 or 30 min. To picture this condition, there is illustrated in Fig. 22 our form of subframe mounting.

The unitary system, if properly carried out, guarantees minimum loss of bus-hours, minimum operating cost, and minimum difficulties from the standpoint of training employes. Obviously, less skill is required on the part of mechanics when they are constantly performing the same operation; here it is simply a question of specialization. But where the construction is such that multi-repair operations are required, the situation is much more complicated. Summing up, to be obliged to remove several units before a faulty unit can be inspected, repaired or replaced, is a condition not to be considered for a moment. Such practice would be ruinous from a public utility standpoint.

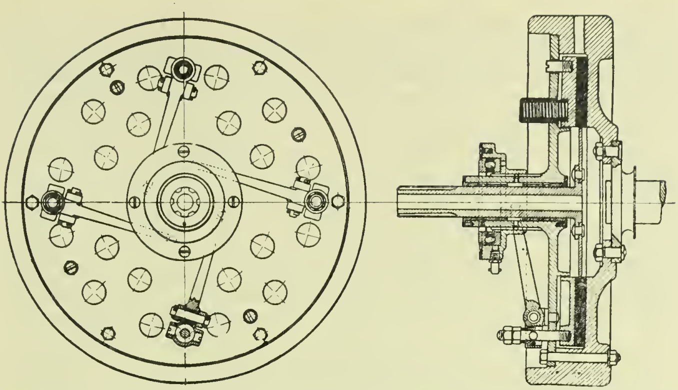


FIG. 21—SECTION THROUGH THE CLUTCH USED ON THE TYPES-J AND L BUSES

It must be remembered that the general conditions surrounding repair work are seldom ideal. There is the matter of wet floors, dirt surrounding the various units, often lack of light. Garage repair forces must work Saturdays and Sundays, which is not particularly attractive. In actual practice it is exceedingly difficult to find men who are willing to work at night. Taken as a whole, the conditions surrounding the work of the repair-men seldom bear favorable comparison with modern high-class factory practice. Here again we wish to emphasize the desirability of unit construction, for the theory is to remove the defective unit and take it to a central repair plant having all the advantages of the modern factory, so that the repairs can be promptly made by skilled men working under the best possible surroundings.

In connection with the matter of accessibility, it should be remembered that repairs and adjustments must be occasionally carried out at night, sometimes under most

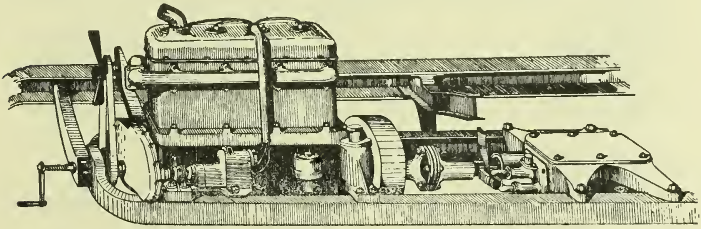


FIG. 22—SUB-FRAME MOUNTING OF THE TYPE-J BUS

unfavorable conditions. Again, assuming the use of low-level equipment, the design should be such that inspections, repairs and renewals can in practically all instances be undertaken from the sides or underneath the vehicles. This means the use of pits. The practice of providing trap-doors inside buses is not desirable. Trap-doors weaken the bodies, are a possible source of accidents, cannot be kept tight in place, permit exhaust gases to leak through, and create undue noise. Experience has shown that it is highly unsatisfactory to carry out chassis repairs from the inside of the body. If this practice is indulged in, claims are bound to result from passengers due to their clothes coming into contact with grease or dirt. Mechanics are sometimes careless and this results in unnecessary damage to the interior fittings, particularly the seat cushions.

MINIMUM CONSUMPTION OF LABOR AND MATERIAL

From a financial viewpoint, the success or failure of a utility operating buses depends upon the cumulative additions or subtractions of small amounts expended on either labor or material. Sometimes the items may appear insignificant but, taken as a whole and over lengthy periods, the story is entirely different. When working, a bus is a heavy consumer of both labor and material. The consumption is perhaps much greater than is generally supposed. To afford a practical illustration, Table 5 representing the actual consumption by our company of some of the major elements for the year 1921, may be of interest. These figures are based on the average of all buses.

From a casual study of these data it will be seen that a relatively small percentage of saving, if applied to any of the items and then multiplied by a large number of

TABLE 5—FIFTH AVENUE COACH CO.'S COST PER BUS FOR 1921

Gasoline			$1,125.94
Lubrication			109.42
Tires			284.34
Repairs to Chassis	Labor	$676.97	1,436.78
	Material	759.81	
Repairs to Bodies	Labor	359.00	521.44
	Material	162.44	
Drivers			3,071.71
Conductors			2,692.48
Total			$9,242.11

vehicles, must total a vast sum annually. If one assumes that the equipment in question is of good design and that its maintenance is economically undertaken, then how much more important does this issue become when the reverse is true.

Perhaps it will not be out of place here to point out that the profit of the average utility expressed in percentage, usually does not run beyond one figure, and that there are a vast number of utilities where the figure is in red. To change the color and to exceed the single-figure basis, requires all that is best in design, material, workmanship and operating care.

MINIMUM CONSUMPTION OF FUEL

Aside from the human elements which have been covered in a previous paper, Motor-Bus Transportation,[3] presented at the 1920 Semi-Annual Meeting, the major issue, of course, is the engine. We employ exclusively the sleeve-valve type. From our viewpoint this type possesses certain basic advantages which make for economy of operation.

First, taking the question of fuel, high gasoline-economy is possible due to

(1) Absence of valve pockets and the spherically shaped combustion-chamber. Incidentally, this permits of high compression being employed. The illustration of the combustion-chamber (Fig. 23) brings out this point very clearly
(2) Positive action of valves at all speeds
(3) Extraordinarily low friction-horsepower
(4) Ideal location of the spark-plug

Next, there is the question of service. In this respect we believe the sleeve-valve engine has the following advantages:

(1) The performance remains reasonably constant throughout the useful life. It is not necessary to make adjustments constantly to permit of satisfactory and uniform behavior
(2) Throughout the useful life the performance tends to improve
(3) Practically no adjustments can be made since there is nothing to adjust. This alone represents a considerable saving in the garage force

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[3] See TRANSACTIONS, vol. 15, part 2, p. 143.

(4) Throughout useful life there is little, if any, increase of noise due to wear

(5) Cost of repairs is small since there are very few operations requiring skill

(6) Cylinders never require reboring. This obviates the necessity of carrying in stock second-standard pistons and rings

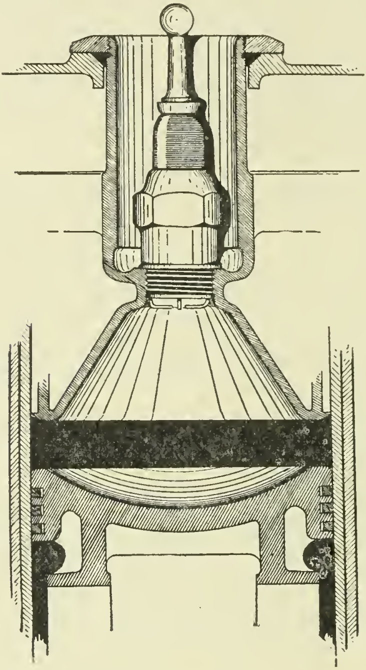

FIG. 23—SECTIONAL ELEVATION THROUGH THE COMBUSTION-CHAMBER OF THE ENGINE USED ON THE TYPES-J AND L BUSES

From Fig. 24 it will be seen that the engine has an exceedingly clean appearance.

The performance of a correctly designed engine is largely a function of its carbureter; therefore a wide variety of results is always obtainable with varied settings. From the graph showing fuel and power output reproduced in Fig. 25 it will be noticed that the characteristics of the sleeve-valve engine are rather remarkable. The setting in question is considered as being particularly suitable for type-J equipment. The points brought out in Table 6 are of special interest.

Expressing the results obtained in another manner, it

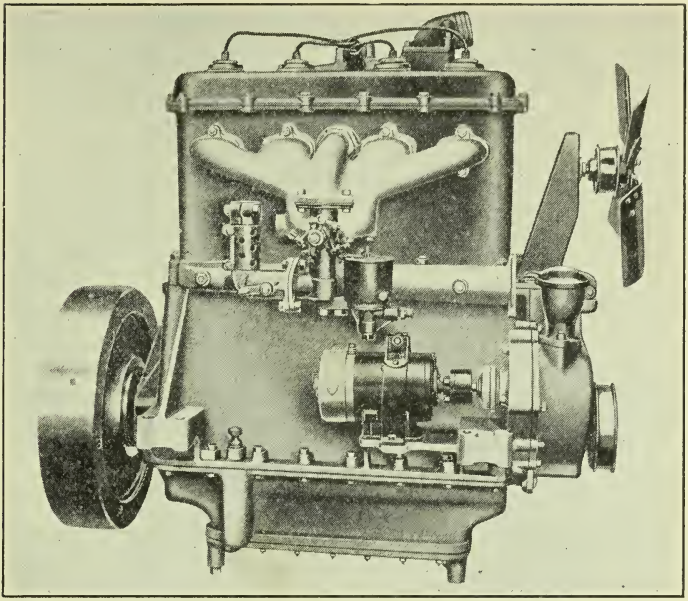

FIG. 24—THE ENGINE USED ON THE TYPES-J AND L BUSES

is interesting to reflect on the fact that during 1921 our entire fleet of buses averaged 50.7 ton-miles per gal. In connection with the rather remarkable performance which this type of engine delivers in our service, particularly from the standpoint of fuel economy, mention should be made of the carbureter, which is of the Zenith type. From Fig. 26 it will be seen that there is no exterior adjustment. The throttle spindle is 7/16 in. in diameter, hardened and ground. There is a total of 4 in. spindle

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TABLE 6—HORSEPOWER AND TORQUE DATA FOR TYPE-J BUS

Power Developed at 1,000 R.P.M., hp.	36.20
Power Developed per Cubic Inch of Displacement, hp.	0.12
Weight of Vehicle per Horsepower, lb.	301.00
Weight of Vehicle per Cubic Inch of Displacement, lb.	36.20
Maximum Torque, lb-ft.	194.00
Speed for Maximum Torque, r.p.m.	800.00
Decrease in Torque at 400 R.P.M., per cent	5.10
Decrease in Torque at 1,400 R.P.M., per cent	11.90
Speed for Maximum Torque with a 5.4 to 1 Rear-Axle Ratio, m.p.h.	16.10
Minimum Fuel-Consumption, lb. per b.hp-hr.	0.55

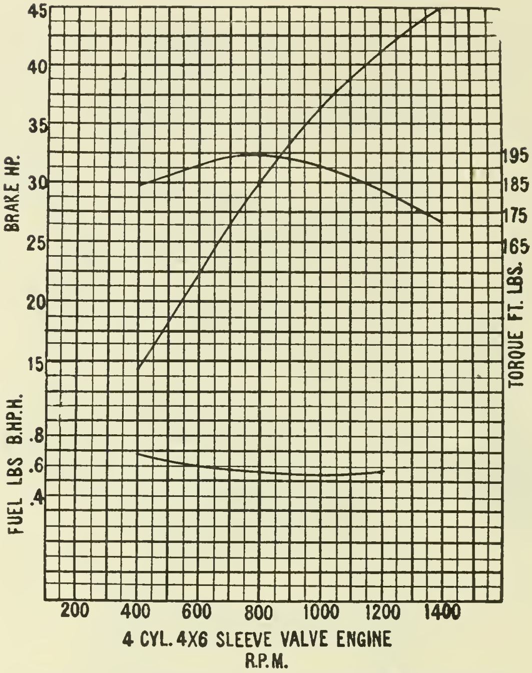

Fig. 25—Curves Showing Engine Performance

bearing-area. There is a gland with a suitable packing at the front end and a blank nut at the other. It is interesting to compare the arrangement with conventional designs that in many instances have throttle spindles resembling closely wire nails. With the bus there is an abnormal amount of throttle movement, and unless this factor is taken into consideration from the standpoint of design, rapid spindle and bearing wear will take place. It will also be seen that the design is rugged throughout. All screws, nuts, plugs or unions are of ample size. The butterfly is exceedingly well fitted and provision is made for a simple throttle-stop adjustment. These points are clearly brought out by Fig. 26.

Minimum Weight

It seems scarcely necessary here to argue as to the desirability of light weight. These remarks particularly apply to the matter of unsprung weight. Assuming good design, obviously minimum weight means minimum fuel-consumption, maximum acceleration and speed and minimum costs for repairs and renewals. These are the controlling elements. Henry Ford started out with this idea firmly imbedded in his mind and, so far as we know, he has had no cause to change his views.

Clearly, the lighter the vehicle, the easier the solution of our problems. Heavy vehicle-weight means unnecessarily large tires, stronger axles and frame, larger brakes, slower gear-ratios and, last but not least, more engine power. The entire theory of design should be based on the highest safe vehicle-speed for the smallest throttle-opening, and consequently the minimum number of engine revolutions. Of course, this is out of the question if we start off with an unnecessarily heavy unit.

From our experience in operating 21 different types of buses in the last 14 years, we believe that the weights and percentages of axle-load distribution given in Table 7 make for safe and efficient practice.

MAXIMUM SAFE SPEED

The greatest single factor from the standpoint of economical operation is speed. This point is perhaps not sufficiently recognized. The following facts in connection with our operation may make the matter somewhat clearer. During 1921 we spent in platform payment, drivers' and conductors' wages, in round figures, $1,625,-

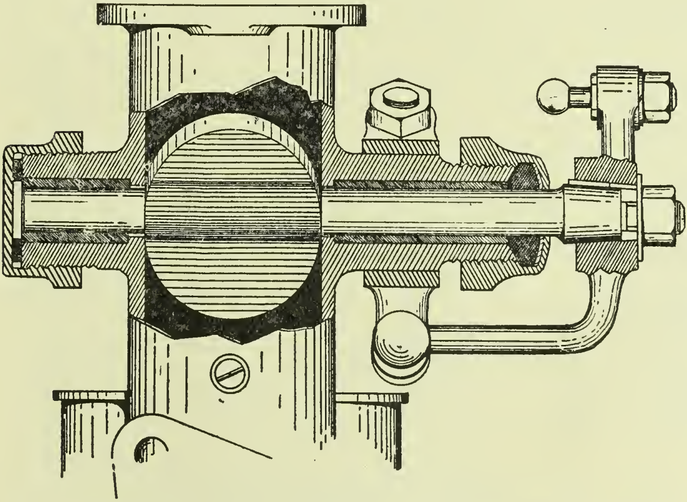

FIG. 26—SECTIONAL VIEW THROUGH THE CARBURETER THROTTLE OF THE TYPES-J AND L ENGINE

TABLE 7—DISTRIBUTION OF WEIGHT IN 51-PASSENGER DOUBLE-DECK AND 25-PASSENGER SINGLE-DECK BUSES

Description	Double-Deck Bus					Single-Deck Bus				
	Total, Lb.	Front		Rear		Total, Lb.	Front		Rear	
		Lb.	Per Cent	Lb.	Per Cent		Lb.	Per Cent	Lb.	Per Cent
Chassis, Gasoline, Oil, Water and Foot-Boards	6,000	3,000	50	3,000	50	5,000	2,600	52	2,400	48
Body, Sign, Battery and Heaters	4,000	880	22	3,120	78	2,000	500	25	1,500	75
Chassis and Body	10,000	3,880	39	6,120	61	7,000	3,100	44	3,900	56
Passengers and Crew at 150 Lb. Each	7,950	2,385	30	5,565	70	3,900	430	11	3,470	89
Grand Total of Fully Loaded Bus	17,950	6,265	35	11,685	65	10,900	3,530	33	7,370	67
Load on Each Tire	....	3,133[a]	....	5,842[b]	....	....	1,765[a]	....	3,685[c]	....
Unsprung Weight, Springs to Tires	....	700	....	2,000	....	....	650	....	1,550	....
Chassis Load on Each Spring	....	1,150	....	500	....	....	975	....	425	....
Body Load on Each Spring	....	440	....	1,560	....	....	250	....	750	....
Passenger and Crew Load on Each Spring	....	1,193	....	2,782	....	....	215	....	1,735	....
Total Load on Each Spring	....	2,783	....	4,842	....	....	1,440	....	2,910	....

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[a] Size 34 x 4 in. single.

[b] Size 34 x 5 in. dual. Figure is for each set.

[c] Size 34 x 6 in. single.

000. So, for each 1-per cent economy in speed there is a yearly potential saving of more than $16,000. Looking at the situation another way, the ratio of expenditure between our platform payment and all money expended in connection with repairs and renewals to chassis and bodies is approximately 5 to 1.

From this it is clear that, while there are always opportunities to effect a saving in connection with maintenance methods generally, the real solution is to employ the fastest possible safe speed and to drive the vehicles up to the limit of their endurance. This, of course, necessitates all that is best from the standpoint of design. Naturally, to maintain a high average rate of speed, rapid acceleration is essential. But in connection with this matter it is well to bear in mind that there is nothing gained and much lost if the engine power is in excess of actual requirements, for it is bound to be abused. A very real problem is to ascertain with each operation the exact amount of power required, then to adopt a standard carbureter-setting with a view to its proper control. Obviously, the questions of acceleration, deceleration and maximum safe speed are closely allied. Reference has been made to deceleration under the heading Effective Brakes.

Maximum Tire-Mileage

In the earlier days of bus operation, the tire question was one of our chief anxieties. Today the situation is very different, for wonderful improvements have been made in tire manufacturing methods. Of course, there is no sense in decreasing tire expenditures at the cost of the equipment generally. Resilient tires are essential and too great a wear must not be permitted. It is our regular practice to remove a tire immediately the rubber has worn to within ⅞ in. of the hard base.

In looking back over our records, it is extremely interesting to note that in 1911 our cost per mile for tires was 4.93 cents. From that date on, a steady reduction has been effected. The figure for 1921 was 0.87 cents per mile, and this, of course, includes the use of six tires. From our viewpoint the factors which have permitted this condition to be reached are, in the order of their importance

(1) Better tire manufacturing methods

(2) Improved vehicle design. This includes decreased

weight, particularly unsprung weight, the substitution of metal for wood wheels, etc.

(3) Closer supervision from an operating standpoint

(4) Closer supervision from a maintenance standpoint

CONCLUSION

As the result of long experience in connection with the design, construction and operation of buses, we are convinced more than ever that trucks or automobiles, modified or unmodified, are absolutely incapable of giving satisfactory and economical service if operated as buses. The tendency today is to employ trucks or automobile chassis as buses, or to attempt to modify their construction, then to re-christen them. This is a dangerous policy from the standpoint of both the builder and the user, and eventually it must surely result in dissatisfaction and disillusionment.

There is another and very important matter: We must not lose sight of the fact that the bus has not made good in some of the localities where it has been tried out. We are constantly confronted with failures such as those at Des Moines, Toledo, Kansas City, and other cities. Such failures, when analyzed, invariably point to the fact that the combination of extemporized equipment, indiscriminate operation, overloading and lack of experience is responsible. But these failures can be avoided, and the automotive industry in its own interest should do all that is possible to guard against such occurrences.

It seems scarcely necessary here to comment upon the splendid achievements of the Society in connection with standardization work in general. Certainly, this has been a controlling influence in the development of the automotive industry. We believe much would be gained if it should now concentrate upon the motorbus. What we have in mind is the standardization of certain of the main dimensions; for example, front and rear-axle tracks, spring center-to-center distances, frame width, dimension between dash and wheel pocket, seat dimensions, aisle widths, etc., for the various classes of service.

The main object of this paper is to bring to the attention of interested parties in a clearcut, vigorous and interesting manner, the fact that to produce motorbus chassis that can be operated efficiently and economically, a very close study must be made of the entire situation. It is also desired to destroy so far as possible the illusion that a bus chassis is merely a modified truck. If in these things, even a moderate degree of success is achieved, we shall feel amply repaid for our efforts.

The matter of body design has not been touched upon, since this is a subject that, because of its magnitude, must receive separate treatment.

THE DISCUSSION

GEORGE W. CRAVENS:—Some years ago I had charge of locomotive design for the General Electric Co. Among the locomotives we designed were the first ones for the New York Central Railroad, which were made with the motors as the frames of the trucks. The center of gravity was very low; in fact, most of the weight was in the trucks and much of it was unsprung. The result was that the first time the New York Central locomotive tried to make high speed on some of the curves it tipped over, due to the fact that the rails tipped over because the center of gravity was so low and the weight mostly unsprung. I state that first because I want to ask Mr. Green if any of his experiments showed that the center of gravity might be too low for safety and for comfort, but especially from the standpoint of safety due to putting excessive side-strains on the wheels when striking obstructions in going around curves. I notice that he favors the steel wheel. I want to raise the question as to the effect of low center of gravity throwing excessive strains on the wheels, causing side-strains on the bearings and springs, and what effect that has upon the riding qualities as well as on the maintenance of the running-gear.

For interurban bus-transportation, has the double-deck bus ever been used successfully? Is it better on general principles outside of cities, where turning radius is not important, to use a short wheelbase with a double-deck or a longer wheelbase with a single-deck bus? About what does experience show as a fair average seating-capacity that should be satisfactory for the purpose?

HARRY A. TARANTOUS:—In Table 5 Mr. Green mentions a material charge of $759.81 for chassis repairs. Is there any cost for sleeve replacement during the year included in that? Is any special grade of gasoline used, or is a standard gasoline sold in New York City employed?

B. S. PFEIFFER:—Is there any difference in the gasoline consumption per ton-mile of the J-type and the L-type buses?

MERRILL C. HORINE:—There has been some talk of the danger of excessive overhang back of the real axle, because of the necessity of turning in congested thoroughfares. Is that really a factor and is there a possibility of side-swiping a neighboring vehicle on the turn?

It is a popular notion that the outside stairway is a danger, because of the possibilities of the passenger falling off. I myself have had to hold on pretty tightly in making stops on Fifth Avenue buses, but I have never fallen off. Did anyone ever fall off?

A very low frame and floor with ordinary-size wheels requires an exceedingly large wheelhouse which, in the case of some buses, entirely eliminates leg-room from at least two seats in the bus. Is there any cure for that, if it is found to be a real disadvantage in practical operation?

In connection with the flat floor Mr. Green advocates in the driver's compartment, that means that the driver has no toe-board. Does not that require the driver to ride the clutch continuously; if so, is there any objection to that if the clutch throw-out bearing is properly designed?

With further reference to the clutch, does Mr. Green favor the use of a clutch brake? I had not noticed much clutch-brake effect in shifting gears on Fifth Avenue buses, and I wondered if there is any reason for that.

I notice that the frames are straight in Mr. Green's designs. There must be a reason why he has not gone to the rather radical drop and gooseneck frames that we are beginning to see on some of the special bus designs.

T. V. BUCKWALTER:—On the question of height of center of gravity, Mr. Cravens has explained why they found it desirable to build a high-center-of-gravity New York Central locomotive, and asked whether that applies to our automotive practice. The first electric locomotives on the New York Central were built with a very low center of gravity and the motors were built on the axle. A rather small wheel for a locomotive, about 44 in. in diameter, was used, and it was found that the riding characteristics were very bad. The locomotive had an exceptionally severe surge on the rail; when the wheels struck a curve, they would hit the rail with a series of jerks. The rails would not stand up under this surging. I think it has been corrected by the redesign of the locomotive to use trailer trucks which ease the locomotive in going around curves.

Before the Pennsylvania Railroad Co. put its locomotives into service in 1910 at the time of the New York electrification, it built a series of Brinell testing machines, four of which were put on each tie, and these testing machines were spaced along about 1 mile of track. They tried out all the various kinds of electric locomotive they could obtain, in comparison with steam locomotives. After a high-speed run they would take the Brinell reading in horizontal and vertical directions on each one of these ties for possibly 1 mile of track, an expensive and laborious operation but well worth while in developing where the center of gravity should be. It developed from this test that the low-center-of-gravity locomotive had the most destructive effect on the roadbed and, as the other types of locomotive were tried out and the center of gravity brought up higher, as I recall it to between 70 and 80 in., the destructive effect on the roadbed and likewise on the locomotive was lessened, until they reached the old type of high-speed passenger-locomotive with 80-in. drive-wheels and the boiler set on top of the frame. One of the earliest of the high-up locomotives with a clear space under the boiler over the wheels was the easiest riding locomotive of them all; and it was the only one that attained a speed of 107 m.p.h.

To apply this to the automotive situation, I am inclined to believe the reason the low center of gravity is so destructive in railroad work is that the surging must be resisted by one set of flanges, the flanges on the outside of a curve; the inside set of flanges has no function in keeping the equipment on the track. The reverse is true in automotive practice. We have four flanges, flanges on both sides. The adhesion of the tires to the ground constitutes the flange action and, by keeping the center of gravity fairly low, we get the effect of double-flange action. In other words, the outside wheels possibly have increased adhesion due to the increased weight that is due to centrifugal force; their effect in keeping the vehicle on its road is reinforced by the weight remaining on the inside wheels. Therefore, I think it is desirable that the center of gravity be kept fairly low.

In regard to the future of the bus as compared with the electric railway, the railway men saw the electrical manufacturing companies practically develop an electric locomotive. One of the reasons for that is that the railway men are so set in their ways that they cannot get into the habit of thinking originally on a new line of work. Mr. Green has given us a fine example of original thought on a new problem. He has thrown into the discard the practices that he has seen on other lines of automotive apparatus and has attacked this problem with a clean slate, working out each of the various details as a problem to be attacked from the ground up. That his solution has been wonderfully successful, as it has, indicates that this is the proper procedure by which to handle a problem of that nature.

The railway men have not been inclined to do that. When the first rail-car bus came up for discussion, some of the earlier specifications were to the effect that it must have Master Car Builders' trucks, a buffing resistance through the center sills of 550,000 lb. and the like. All of that applied 10 to 15 years ago to the gasoline rail-car and, at that time, they would not permit us to disregard those features. The cars built at that time could not be anything but failures; a problem of that kind must be attacked as a new problem and we must forget all our ancient theory.

I feel that there is a great future for the motorbus because the bus can receive and discharge its passengers at the curb and leave the center of the street free for other traffic. I believe that this one feature alone will increase the capacity of our streets in larger cities at least 100 per cent; that is, double the traffic over a street that has electric railway equipment, which in a good many of our crowded communities, where the streets are not wide, requires the stopping and holding-up of all automotive traffic because of the presence of the track-confined vehicles.

The second great advantage of the bus consists of the facility with which additional buses can run by one that is stopping to receive or discharge passengers; whereas, in comparison, the track-confined vehicle holds-up all following traffic when stopping for any cause. In like manner, when any trouble does develop in a bus, it can be run up to the curb or into a side street without creating a general hold-up of traffic; whereas it is a common sight to see electric rail-equipment bunched, often due to some minor defect or adjustment that, until corrected, holds-up traffic on that particular track. To sum up, the track-confined vehicle has a general tendency to run in bunches; whereas the tendency of the self-contained bus-system is to distribute traffic over the entire route.

We have endeavored to approach the builders of street-railway equipment with the thought of modernizing some of their equipment. Their general line of defense is that we aim to give our customers what they want. The electric-railway people, on the other hand, say, "We take what the street-railway-equipment company has to offer, and that, because it does not offer certain things, we do not consider them." That vicious cycle is one of the things that prevents the established companies from developing and it is one of the reasons why the gasoline motorbus will succeed ultimately and displace a considerable amount of our electric-railway equipment.

HERBERT CHASE:—To what condition does Mr. Green attribute the unusually low friction that he says is characteristic of the Knight engine? I am aware that his company is using or is planning to use an oil of lower viscosity than is used customarily for that type of engine, but I fail to see why the friction of the Knight engine should be lower than that of an engine of the poppet-valve type.

I am sure that Mr. Green will agree that the bus as it is developed today, even by the Fifth Avenue Coach Co., is not an ideal vehicle in all respects and that there are many possibilities of improvement. It is my belief that these possibilities should be considered; that they should not be overlooked, as they are apt to be, through an effort to apply our present-day type of engine, gearset and the like, to the problem rather than design a vehicle primarily from the standpoint of the service to which it must be applied. The steam-operated bus, for example, has certain inherent advantages, such as high starting-torque, smooth and rapid accelerating ability and the elimination of a gearset. These advantages are realized in the steam buses which have been used in London to some extent for many years. What are Mr. Green's views about the possibilities in this direction?

E. W. WEAVER:—In speaking of the necessity for economy, Mr. Green brought out the fact that a low speed of revolution of the engine per minute and slight opening of the throttle tends to give economy. With the throttle barely cracked the engine has a very low volumetric-efficiency. If it were possible with a different gear-ratio to cut the number of revolutions per minute in two and increase the efficiency, I think the gasoline consumption could be cut down. I am working on a proposition tending in that direction.

A. A. BULL:—In Table 5 chassis repair is all grouped under one heading. Would it be possible for Mr. Green to separate this item so as to indicate in a general way which part of the chassis or component constitutes the greatest item of repair expense?

O. C. BERRY:—I wish to comment on the advantage of the Knight engine as compared with the poppet-valve engine. I am not sure that the Knight engine is superior. The idea that a higher compression can be used in a Knight engine than in a poppet-valve engine is contrary to my experience. Has Mr. Green any special data to bear that out? Secondly, Mr. Green says that the spark-plug is better placed in a Knight engine than in a poppet-valve engine. From my way of looking at it, the spark-plug should be placed at the point where the mixture is richest. In the Knight engine the spark-plug is placed at the point of leanest mixture. This is a matter of considerable importance and seems to offset entirely the other obvious points of advantage. The best figures I have been able to obtain indicate just about equally good performance in the Knight engine and the poppet-valve engine.

HAROLD W. SLAUSON:—Were any tilting tests made to determine the relative stability of the pneumatic and solid tires of the same capacity?

CORNELIUS T. MYERS:—As people gain experience in any industry they accumulate not only a number of positive facts that are very valuable, but certain negative ones. In between the lines of Mr. Green's paper, in which appears the accumulated experience of many years, a lot of "don'ts" will be found. Mr. Green could serve us to a still greater extent if he would put down some of the big "don'ts" that are written down in his book of experience. There are many things that one should not do; these have been tried and found wanting. In that connection, they would be particularly valuable to executives, because executives of street-railway companies and those considering going into bus transportation of any kind must depend upon opinions that at times are not very reliable. The information may be given with all honesty and to the best of the ability of those concerned, but it does not have that reliability that is absolutely essential to the proper development of the motorbus business. The editor of *Railway Age* was inclined recently to be a trifle cynical in viewing the aspects of the rail bus. He said this rail bus appears in cycles of seven years' average and seems to be now in the fourth cycle. I think the discussion of his views showed very clearly that in the previous cycles the railroad men had to depend entirely on their own viewpoints and experience, simply taking the general broad idea of applying a gasoline engine to operating a rail car. If steam-railroad executives and street-railway executives who are considering this matter will call on the experience of the automotive industry, work with it and pay particular attention to the "don'ts" that are written in the books of experience of those who have been in this business, I think we shall have a comparatively rapid and a safe development.

ARTHUR J. SCAIFE:—Mr. Myers' remarks are similar to remarks made during the discussion[4] on the subject of the rail motor-car which was entered into at the meeting of the Metropolitan Section of the Society held in New York City recently. With reference to Mr. Buckwalter's remarks, there was considerable prejudice against the motorbus, and especially on the part of the electric-railway companies. This, however, has disappeared within the last 2 years. The street railways have evidently come to the conclusion that it is much better to get into the bus business than to fight it.

I visited California recently and had opportunity to study the bus business. We have 400 to 500 buses or stages operating in that territory, a number of which are on long-distance work. They are all operating under franchises at present. This business started several years ago in a manner similar to that of a great many of the bus lines that operate today in the Eastern States. On long-distance work the Westerners are, I think, about 3 years ahead of us in the East. A few of the operators have bought up all the small franchises, so there are only about three large operators in California at present.

The buses operate at about 35 m.p.h., maximum. The average wheelbase is 200 in. Each bus carries from 18 to 25 passengers. The later models are equipped with 36 x 6-in. pneumatic front tires and 36 x 6-in. dual rear tires. The rear seat usually is located directly over the rear wheels, similar to passenger-car design. They have a seating space from 32 to 25 in. It is necessary to have ample room for comfort when carrying passengers from 126 to 500 miles. For instance, you can leave San Francisco at 8:00 a. m. and arrive in Los Angeles at 11:00 p. m., on the same stage, using three drivers.

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[4] See page 609.

I took a trip from Los Angeles to Bakersfield, a distance of 126 miles, the trip being made in 5 hr., which is an average speed of 25 m.p.h. This stage carries 18 passengers and has a four-speed transmission, with a maximum road speed of 35 m.p.h. In going to Bakersfield we passed over three mountain ranges, having 1130 curves on the 29 miles of mountain; the maximum grade is 9 per cent and the average grade 6 per cent. They maintain a very good average speed over these ranges and around the curves, but if you are subject to seasickness do not ride in the back seat, as they say that nausea is very common when sitting there.

There is no doubt that motorbuses are coming to stay, but just what the ultimate bus will be for suburban, city or interurban work is a question that will have to be settled within the next 10 years.

R. E. PLIMPTON:—Mr. Green mentions an acceleration of 2 m.p.h. per sec. per sec. I think that applies mainly to buses of the cross-seat type and is too high for those with the longitudinal seats. For the latter 1½ m.p.h. per sec. per sec. is probably the maximum that should be used. At 2 or 2½ m.p.h. per sec. per sec. passengers are thrown toward the driver or to the rear of the vehicle when it is being started or stopped.

Mr. Cravens brought up the matter of double-deck buses for inter-city operation. Even in smaller towns the bus operators often find that the standard single-deck bus gives trouble because of trees or other obstructions. On long-distance work there are likely to be many obstacles of that sort, even though the traffic is sufficiently dense to warrant the double-deck vehicle.

G. A. GREEN:—Mr. Cravens asks if we have ascertained, as a result of our experiments, whether it is possible to reach a point where the center of gravity is too low, since, under such circumstances, excessive stresses would be imposed on wheels, bearings and springs, thus adversely affecting the riding properties and the maintenance costs generally. With the bus, from a low-level standpoint, ground-clearance is the limiting factor. With our L-type bus this has been cut down to 6 in., which we consider closely approaches the minimum. We do not believe a lower center of gravity is either necessary or desirable. With our design the riding properties can be considered good, and, compared with our high-level equipment, there is no appreciable increase in maintenance costs. But vehicles with a low center of gravity do present many serious problems in regard to the side stresses mentioned. These problems are not at all easy of solution.

It is rather interesting to note that the appearance of a vehicle is subject to marked change as its height is decreased. A vehicle lower than our L-type bus gives an impression of a box formation and, consequently, the artistic value of the finished product is greatly lessened.

Answering Mr. Tarantous' questions, the $759.81 covering material used in connection with repairs to the chassis as per Table 5, naturally includes repairs and renewals to sleeves. The renewal of sleeves cannot be considered as particularly serious. They are not very expensive. They should last at least a year and their replacement is a comparatively simple matter.

With regard to the matter of fuel, standard gasoline as sold in New York City is employed exclusively by us. This gasoline meets the Navy Department's specifications.

Referring to Mr. Pfeiffer's query relative to gasoline ton-mileage with our L-type and J-type buses, since the same powerplant is employed in both cases, a marked difference is not to be expected. We do, however, obtain a somewhat better average with the J-type bus. Primarily, this is due to minor refinements in connection with the engine design. With regard to the matter of fuel consumption expressed in terms of miles per gallon, in this respect the J-type bus shows a very marked improvement over the L-type bus. This, of course, is natural because of the lighter vehicle-weight and less number of passengers carried. As a rule, a single-deck bus is expected to accelerate more rapidly than a double-deck bus, and acceleration is very costly from the standpoint of fuel. In this connection it seems likely that the average builder and user of single-deck buses will be somewhat disappointed with his fuel-consumption figures.

In reply to Mr. Horine, a rear overhang of an abnormal length is certainly a hazard. It is clear that such construction necessitates particular skill on the part of the driver to clear obstacles when rounding corners. Another very serious matter is that of riding properties. Naturally, if the wheels are too far from the rear end, the passengers on the rear seats will be subject to considerable movement.

The danger of the outside stairway is more apparent than real. So far as my recollection goes, we have had only one or two accidents due to people falling from the stairway. In view of the fact that we have carried in excess of 300,000,000 passengers, the risk cannot be considered as particularly serious.

Large wheel-housings are unquestionably most unsatisfactory from the passengers' standpoint. Of course, the depth of the housings is automatically decreased due to the wide track and frame, the necessity for which is emphasized in my paper. In short, the outer edge of the tires should closely correspond to the extreme overall width of the body and the shape of the housings should conform as nearly as possible to the adjacent members such as brake-drums and the like. The best proposition appears to be a cast or pressed housing with rounded corners and an integral panel.

Regarding the necessity for a flat floor at the front end, this is desirable primarily to avoid accidents, since obstructions of any kind are liable to interfere with ingoing and outgoing passengers. In any event, the natural position for the driver's feet is in the horizontal plane.

Concerning comments on the clutch, a clutch brake is not employed in our service due to the fact that up to the present we have not been able to develop a satisfactory design. The conventional arrangement, though helpful when changing into higher speeds, is decidedly a deterrent when changing down. It must be admitted that many of our men are not expert at gearshifting; but it will be readily understood that we are extremely anxious to keep to our schedules; in short, to employ the highest possible safe speed. In considering that 1 per cent of lost time represents in round figures $16,000 in the form of wages and $50,000 as receipts annually, it will be recognized at once that some damage to the gearing is not extraordinarily serious. Lastly, it is our aim from the standpoint of design to provide everything that is essential but nothing that can be dispensed with, and the clutch brake naturally falls under this heading.

As to straight versus drop frames, our L-type bus has a kick-up at both the front and the rear; but the J-type bus has this at the rear only. We believe a double kick-up is necessary only for double-deck buses where the frame must be kept extraordinarily low. The double-deck bus is intended for operation on city pavements that are usually in fair condition. The single-deck bus is a utility vehicle; this class of vehicle will have a much wider use and therefore the matter of road-clearance must be taken into account. Of course, in many cases single-deck vehicles will be operated over very bad roads. In view of these facts it would seem bad policy to construct vehicles the frame design of which positively restricts their employment to certain localities.

Replying to Mr. Chase, as a result of many tests carried out to determine friction-horsepower losses in connection with various types of engine, we are satisfied that our Knight-type engine possesses remarkable properties in this respect. Table 8 gives figures representing an average of many tests which bring out this point clearly.

TABLE 8—FRICTION HORSEPOWER

Engine Speed, r.p.m.	Friction Horsepower
600	2.2
700	2.6
800	3.1
900	3.8
1,000	4.5

The fact of the matter is that with the poppet-valve engine a surprisingly large amount of power is consumed in connection with the operation of the valves and valve mechanism. In this connection it is suggested that builders of poppet-valve engines might gain something by studying this question even more closely in the future than they have studied it in the past. The best proposition is to carry out what we term progressive friction-horsepower tests. The thought is to check up electrically the power required to turn the engine over, starting first with the crankshaft in its bearings and then gradually adding the reciprocating parts, such as the valve mechanism, valves and the like.

Regarding the matter of design, insofar as our product is concerned, we can only say that we are building vehicles that have embodied in them many of the principles that we believe are essential to permit of safe, economical and satisfactory operation. Of course, the design cannot be considered as perfect. From a motorbus standpoint the automotive industry is in its very early stages, and any design must of necessity be in some respects out-of-date even before its production could be commenced, assuming of course that there was no delay.

We realize that there are inherent advantages in other types of propulsion. I thoroughly agree with Mr. Chase that steam has wonderful possibilities. An entirely new field is opened in connection with the application of steam along automotive lines for the hauling of light loads on steel roadbeds. This point is drawn particularly to the attention of those who are enthusiastic in connection with the use of gasoline-propelled railroad-units. We recognize the merits of steam, but our entire organization has been trained along gasoline-usage lines. We therefore feel, at least at present, that we should concentrate our efforts on the development of the product with which we are most familiar.

Mr. Weaver has criticized my statement to the effect that the entire theory of design should be based on the highest safe vehicle-speed for the smallest throttle-opening and consequent minimum number of engine revolutions. His contention is that, under such conditions, the engine has a low volumetric-efficiency. Substantially Mr. Weaver is correct and possibly my remarks are somewhat misleading, but they are not intended to be taken too literally. As a result, in our service the throttles are sufficiently open to avoid the condition mentioned. About the only time they are almost closed is when the vehicles are at rest and the engines idling.

Mr. Bull has requested that a further analysis be made of Table 5 of my paper. He desires that the item of $759.81 for material used in repairs to chassis be subdivided, preferably under the various units. Obviously this will demand a considerable amount of study. In view of this, perhaps it will be considered sufficient to indicate the units in order of their importance from the standpoint of renewal and repair cost, as follows:

(1) Engine
(2) Transmission
(3) Brakes
(4) Clutch
(5) Rear Axle
(6) Front Axle
(7) Steering
(8) Radiator

Mr. Berry's queries are answered as follows: Regarding our contention that a higher compression can be used with the sleeve-type than with the poppet-valve engine, we feel this is due primarily to the absence of valve-pockets and the spherically shaped combustion-chamber. Our statements are based on numerous comparative tests that have been carried out in connection with our own service.

Regarding the center spark-plug location, this may be incorrect theoretically, but we doubt it. As a matter of fact, we have accumulated an almost unique record from the standpoint of fuel economy. Our figures show consistent economy and are superior to anything we have been able to obtain with poppet-valve engines, regardless of the spark-plug location.

As to the matter of fuel economy, we average in excess of 7 miles per gal. with our double-deck buses throughout the year. With the single-deck vehicles we average from 10 to 11 miles per gal. Recently we sent a J-type bus to Philadelphia on a test run. The fuel average was 14½ miles per gal.; the speed average 22½ m.p.h. It was not necessary to add any water, and but 2 pints of oil was consumed.

If I understand Mr. Slauson's question correctly, he has asked whether we have made any tilting tests to compare the relative stability of vehicles equipped with pneumatic, semi-pneumatic or solid tires. All our tilting tests have been made with solid tires, but, of course, assuming wheels having the same diameter, we would not expect to find any difference in regard to stability.

The wisdom of preparing a series of "don'ts" such as was outlined by Mr. Myers appears somewhat debatable. Those interested in the construction or operation of buses unquestionably will be able to read between the lines of my paper, and sufficient information has been furnished to enable the acceptance or rejection of the various theories propounded. The power of suggestion is the thing that really counts and, according to my viewpoint, no man, regardless of his standing in the industry, should have the temerity to state definitely before the Society what it should or should not do. He should simply give his views, or, better still, the views of his organization and the reasons underlying them.

As to the utility of various forms of equipment, it would seem that any operation requires two types. In the majority of instances initial operation can profitably be commenced with single-deck vehicles; then, after the service is built up, double-deck buses should be added. Even after the service has reached a point where double-deck vehicles are required, the single-deck type will be found extremely valuable in aiding the natural process of development and for operating during cold and wet weather, or for all kinds of special service.

FUNDAMENTAL CHARACTERISTICS OF PRESENT-DAY BUSES[1]

By R E Plimpton[2]

The author enumerates the distinctive features of buses designed for city, for inter-city and for country service and comments upon them, presenting illustrations of these types of bus. Steam and electric motive power are discussed and the chassis components for bus service are considered in some detail. The general types of bus body are treated, together with the influences of climatic conditions and local preferences.

Comfort and convenience factors are discussed at some length and the problems of heating, lighting and ventilation are given constructive attention. Fare-collection devices and methods are commented upon, and the State and local legal regulations are referred to in connection with their effect upon bus operation.

Illustrations are included and a table showing condensed specifications for city buses is presented.

Any division of the types of vehicle used for buses must be arbitrary, but for convenience we can consider them as applied in city, inter-city and country service. The city bus is designed for use on good pavements. The seating capacity is high, 20 or more passengers, but high speed is not essential. A maximum of 20 m.p.h. is often considered ample. Ability to thread through traffic and to move passengers quickly, on and off, is required. Ordinarily, overload capacity should be provided, so that standees can be accommodated during the rush hours.

The double-decker, a dense-traffic vehicle, has been developed highly through years of service in London and New York. Table 1 gives specifications for the latest types used in these cities. The L-type bus of the Fifth Avenue Coach Co. is distinguished by the low floor, the first step being at the curb and only one other being required to reach the floor of the bus. Increased seating capacity, without any increase of weight, has been se-

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[1] Semi-Annual Meeting paper.

[2] M.S.A.E.—Associate editor, *Bus Transportation*, New York City.

Supplement to Fundamental Characteristics of Present-Day Buses by R. E. Plimpton

TABLE 1—CONDENSED SPECIFICATIONS OF TYPICAL CITY BUSES[3]

Unit	London General Omnibus Co., Type S	Fifth Avenue Coach Co., Type L	Imperial	Fifth Avenue Coach Co., Type J	Mack	Republic	White	Duplex	Fageol
Number of Seats..	54	51	30	25	25	25	25	23	20
Approximate Floor Height at Entrance, in....	31	18	26½	25	32	28	28	27	18½
Wheelbase, in....	179	175	195	172	180	168	198	160	188
Front Gage, in...	72½	67	66½	68	58¼	66	58½	56	70
Rear Gage, in....	70	72	71	71¼	60	66	60¾	62	70
Total Weight, lb..	9,380	10,150	9,500	6,900	8,850	7,900	8,900	6,500	6,800
Body Weight Only, lb........	2,800	4,000	3,000	2,100	2,700	2,400	3,500	2,300	1,600
Engine, Bore and Stroke, in.......	4.26x5.51	4x6	4⅛x5¼	4x6	4¼x5	4⅛x5½	4¼x5¾	4x5¼	4¼x5½
Type of Valves...	L-Head	Sleeve	I-Head	Sleeve	L-Head	Sleeve	L-Head	L-Head	I-Head
Engine Speed, r. p. m.........	1,050	1,400	1,600	1,400	1,275	1,800	1,650	1,600	1,800
Horsepower......	34	50	47½	50	37	50	50	42	62
Fuel Tank Capacity, gal........	25	40	30	35	25	30	35	25	28
Fuel Feed.......	Vacuum	Gravity	Vacuum	Vacuum	Vacuum	Vacuum	Vacuum	Vacuum	Vacuum
Carbureter.......	Zenith	Zenith Special	Zenith	Zenith Special	Schebler	Float-Feed	Own	Stromberg	Zenith
Cooling System...	Pump. Tubular Radiator	Thermo. Tubular Radiator	Pump. Tubular Radiator	Thermo. Tubular Radiator	Pump. Cellular Radiator	Thermo. Tubular Radiator	Pump. Cellular Radiator	Pump. Cellular Radiator	Pump. Cellular Radiator
Ignition.........	Magneto	Magneto	Magneto	Magneto	Magneto	Magneto	Magneto	Battery	Battery
Clutch..........	Single Dry Plate	Single Dry Plate	Multiple, Dry Disc	Single Dry Plate	Multiple Dry Disc	Multiple Dry Disc	Single Plate	Multiple Dry Disc	Multiple Dry Disc
Transmission Speeds.........	3	4	4	4	4	4	4	4	4
Control.........	Center	Side	Center	Side	Center	Center	Center	Center	Center
Universal-Joints..	Fabric and Metal	Fabric	Metal	Fabric	Metal	Metal	Metal	Metal	Metal and Fabric
Steering Gear....	Worm and Nut	Worm and Nut	Screw and Nut	Worm and Nut	Worm and Gear	Worm and Nut	Worm and Sector	Worm and Nut	Worm and Nut
Front Axle.......	I-Beam Section	I-Beam Section	I-Beam Section, Dropped Center	I-Beam Section	I-Beam Section	I-Beam Section	I-Beam Section	I-Beam Section	I-Beam Section
Rear Axle.......	Underslung Worm	Internal Gear	Internal Gear	Worm, Semi-Floating	Double Reduction	Internal Gears	Double Reduction	Worm	Underslung Worm
Wheels..........	Cast Steel	Steel Disc	Steel Disc	Hollow Spoke	Artillery	Artillery Cushion	Cast Steel	Artillery	Disc
Front Tires......	Solid, 41.30x4.73	Solid, 34x5	Cushion, 36x6	Cushion, 34x4	Cushion, 34x5	Cushion, 34x4	Solid, 36x4	Pneumatic, 35x5	Pneumatic 36x6
Rear Tires.......	Dual Solid, 41.30x4.70	Dual Solid, 34x5	Dual Cushion, 36x6	Cushion, 34x6	Dual Cushion, 34x5	Cushion, 34x8	Solid, 36x7	Pneumatic, 38x7	Pneumatic, 36x6
Spring-Suspension	Semi-Elliptical, Volute Auxiliary	Semi-Elliptical, Progressive	Semi-Elliptical, Compensating Underslung	Semi-Elliptical, Progressive	Semi-Elliptical, Rubber Blocks	Semi-Elliptical	Semi-Elliptical	Semi-Elliptical, Snubbers	Semi-Elliptical, Underslung
Braking System..	Two, on Rear Wheels	Drive Shaft and Rear Wheels	Drive Shaft and Rear Wheels	Two, Expanding on Rear Wheels	Drive Shaft and Rear Wheels	Drive Shaft and Rear Wheels	Drive Shaft and Rear Wheels	Two, Expanding on Rear Wheels	Two, Expanding on Rear Wheels

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[3] The first two are double-deck buses; the remainder are single-deck buses.

Supplement to Fundamental Characteristics of Present-Day Buses by R. E. Plimpton

TABLE 1—CONDENSED SPECIFICATIONS OF TYPICAL CITY BUSES[3]

Unit	London General Omnibus	Fifth Avenue Coach	Imperial	Fifth Avenue Coach	Mack	Republic	White	Duplex	Fageol
Braking System . .	Two, on Rear Wheels	Drive Shaft and Rear Wheels	Drive Shaft and Rear Wheels	Two, Expanding on Rear Wheels	Drive Shaft and Rear Wheels	Drive Shaft and Rear Wheels	Drive Shaft and Rear Wheels	Two, Expanding on Rear Wheels	Two, Expanding on Rear Wheels

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[3] The first two are double-deck buses; the remainder are single-deck buses.

cured in many recent double-deckers. A sample now operated in Chicago seats 69 passengers, with a total vehicle weight of only 157 lb. per seat. Previously, about 200 lb. per seat had been considered good practice. This Chicago bus is probably the largest double-decker ever operated commercially. Fig. 1 shows the plan of the lower deck of what is said to be the largest double-decker in England, a Leyland 63-seater carrying 34 passengers on top. Notice the space for a loading well at the lower entrance.

Covered upper decks have been tried out in New York City and Chicago, but have proved popular only in bad weather. Double-deckers in London, England, and Toronto, Canada, have been fitted with blankets for each top-deck seat, in the effort to build up the load factor, these being illustrated in Fig. 2. The so-called double-single-decker has been tried in England; the entire top deck of this vehicle can be detached for winter service, the stairs being removed and the hatchway closed by a trapdoor. One vehicle of this type has a capacity of 29 passengers as a single-decker, and of 54 passengers, half below and half above, when both decks are in use. However, the single-decker shown in Fig. 3 has by far the largest field as a city bus. A few of these designs are described in Table 1 and there are many others of a similar type under way by other builders. Long wheelbase to give loading capacity, a wide rear gage and a narrower gage on the front to give stability and a small turning-circle, low frame-height to give quick passenger movement as well as stability, long springs and special devices for easy riding are the more important features of what promises to be a design that will be used widely in and about our cities.

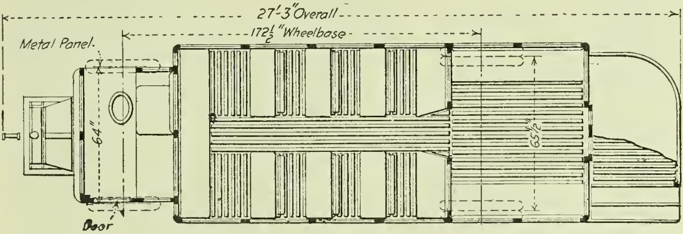

Fig. 1—Plan of the Lower Deck of an English Double-Deck Bus Carrying 63 Passengers

The design for a light, fast, single-deck, one-man bus is being considered for city service by the engineering association of the American Electric Railway Association. It is of interest to know that experts on city transportation favor three sizes, for 21, 25 and 29 passengers, with a maximum floor-height of 26 in. A smaller size, for 16 passengers, has also been discussed.

Most city buses have a central aisle, with cross-seats on each side. In a modified form the seats near the service door are of the longitudinal type, thus providing a space for standees or for those making short trips. This is being applied in short-haul work, while there is a tendency to use the unmodified cross-seat type in suburban work, and even for inter-city service.

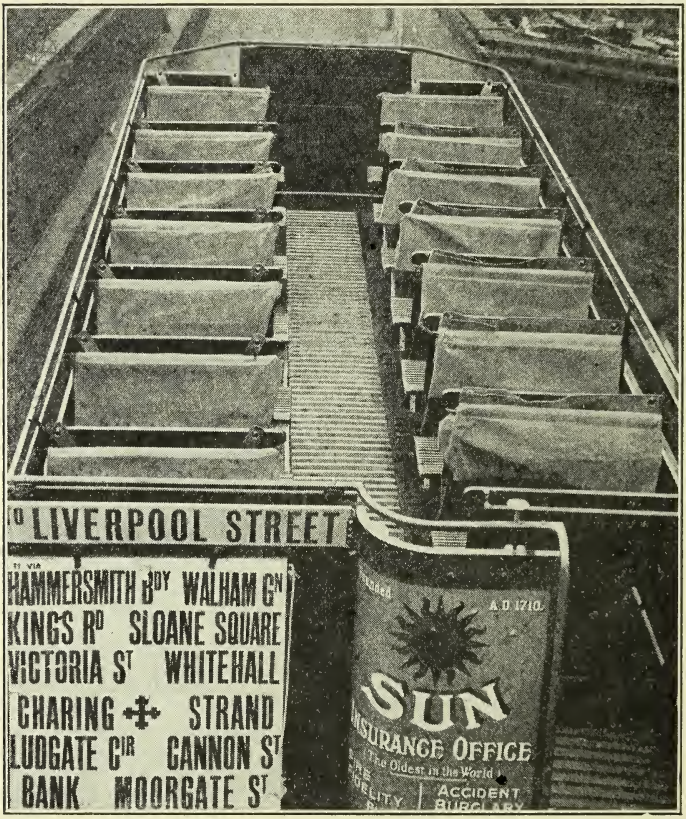
LIVERPOOL STREET
HAMMERSMITH B^DY WALHAM G^N
KINGS R^D SLOANE SQUARE
VICTORIA S^T WHITEHALL
CHARING ✠ STRAND
LUDGATE C^IR CANNON S^T
BANK MOORGATE S^T
A.D. 1710.
SUN
INSURANCE OFFICE
The Oldest in the World
ACCIDENT
BURGLARY

FIG. 2—TARPAULIN LAP ROBES USED ON THE UPPER DECK OF THE LONDON BUSES

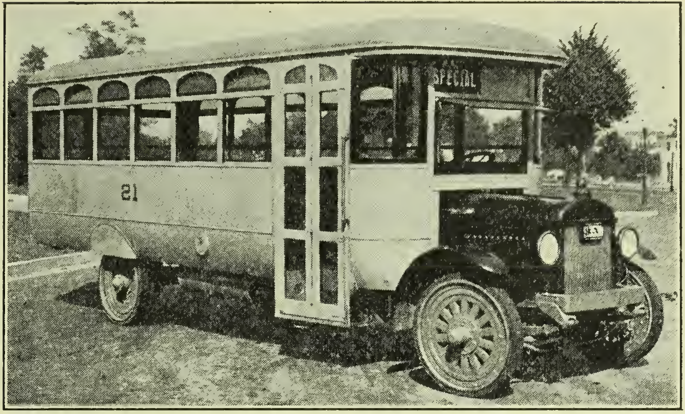

FIG. 3—SINGLE-DECK, 25-PASSENGER BUS FOR CITY SERVICE

THE INTER-CITY BUS

The inter-city bus, such as is shown in Fig. 4, is smaller and faster than the city bus. From 12 to 20 seats are provided for the long runs of 20 miles and upward. These vehicles are known as stages in the West and particularly in California, probably because of familiarity with the term to designate a horse-drawn omnibus. The stage usually is built with three or four rows of seats running all the way across the body, each having its own side doors. It resembles an enlarged touring car and, of course, that is what it frequently is. Comfortable riding is desired, rather than the ability to handle crowds. The floor level is kept as low as possible without decreasing the road clearance below 7 in. The center of gravity is kept down to eliminate sway, with its consequent danger, and undue wear on tires and wheel bearings.

The average speed in Western inter-city service is high. Orders to drivers on a large system in California are to make 32 m.p.h. on well-paved highways and 35 m.p.h. is the legal speed limit. However, there are stretches where it is hard to keep down to this speed. For example, on the desert floor west of El Centro, Cal., a broad concrete pavement stretches away for 30 miles. There are no cross-roads and no habitations are in sight; there is no obstruction to view and very little traffic. With a high-powered car, the driver who will keep down to 35 m.p.h. is unusual. The actual speed over such

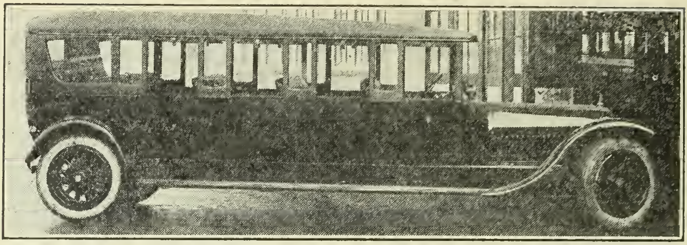

FIG. 4—CALIFORNIA STAGE ON A REBUILT AUTOMOBILE CHASSIS HAVING A CAPACITY OF 14 PASSENGERS ON FIVE CROSS-SEATS

stretches is from 40 to 45 m.p.h. and, occasionally, 50 m.p.h. The chassis and body should be of the minimum weight, consistent with long life and durability. The improvements being made in highway systems will, it is believed, lead to the use of even lighter chassis than are used at present.

The stage used in the West is, as a rule, a rebuilt product. A high-grade touring car is bought second-hand, the frame is lengthened 4 to 5 ft., and a heavier differential, heavier springs and heavier tires are installed. It is realized that these vehicles are expensive, in both first and maintenance costs, since they are always in the "special" class; but Western operators have used them because they could not buy a chassis that would carry the long bodies up steep grades and at high speeds demanded by Western operating conditions. One design at least, that is included in Table 1, is now on the market, and there are indications that others will be available soon for inter-city service, not only on the Pacific coast, but also in many other parts of the Country where the large rebuilt touring car has been used heretofore.

THE COUNTRY BUS

The country bus may start nowhere and end nowhere, as is said of some of them, but usually it runs through rural districts to a trading center or a point where connections can be made with some other form of transportation. This service does not require particularly high speed, but the bus should be able to keep going over poor roads, even when they are covered with mud or snow. Travel is light in this service; so the vehicles used are small. Touring cars of five and seven-passenger capacity may be operated when the roads are particularly bad, giving way to larger buses, up to 18-passenger capacity, during the summer months. One trip a day is often the custom, to town in the morning and returning in the middle of the afternoon. Thus, the operator has the time to pick up light freight for delivery along his route; or he may be a star-route contractor for the Post Office Department, in which case he may carry passengers and freight so long as his contract is properly performed. Therefore, the country bus should be built to carry both passengers and freight. Most operators seem to use the standard passenger vehicle, car or bus, and carry the packages inside or outside, wherever there is vacant space. A space for packages is placed at the front of the vehicle in some designs, with the rear given over to seats.

A convertible vehicle would be particularly useful in consolidated school service, where many buses are now operated only 3 or 4 hr. per day. The seats are removable in one design, so that the school bus can be changed quickly to a light delivery truck. Several combination bodies are on the market, designed to carry both passengers and express, but they have not been used to any extent by bus operators so far.

A transferable body has been suggested for school use, although it might be applied by the operator of country buses also. This would be mounted on a motor-vehicle chassis when the roads were good, but transferred to a horse-drawn vehicle, on wheels or runners, when they are impassable. The fact that this has been proposed seriously indicates the need for properly designed country buses. The field for them is enormous, and it should be possible to provide operators with a light, sturdy, economical bus for use during the 12 months of the year. Some engineers contend that good roads must come first. Yet bus service is being given on thousands of miles of unimproved roads. The life of the vehicles used is short, and some operators believe it economical to turn them in yearly for new ones. Here is a real problem waiting for automotive engineers to solve. The bus for city service is receiving attention from a large number of sources, but its country cousin is, as yet, almost neglected.

STEAM AND ELECTRIC MOTIVE POWER

Before taking up the details of chassis and body used for the standard gasoline bus, other types that have been applied will be noticed. The gasoline-electric bus shown in Fig. 5 is used commercially in England, some 175 of

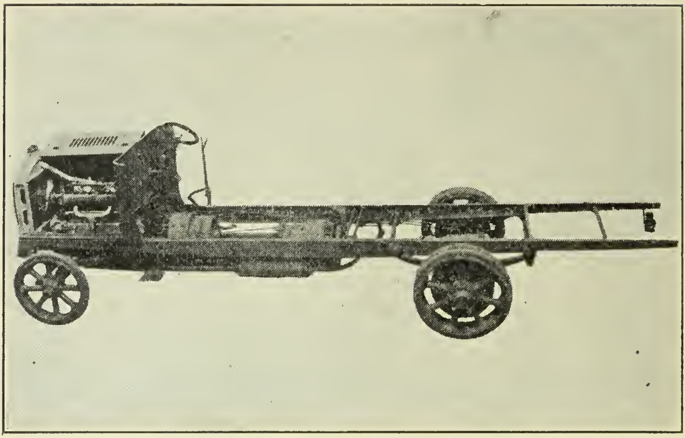

FIG. 5—CHASSIS OF THE TILLING-STEVENS GASOLINE-ELECTRIC BUS USED IN ENGLAND

the Tilling-Stevens chassis being operated in London, and others in the smaller towns. This consists of a gasoline engine direct-connected to a dynamo. The latter is electrically connected to an electric motor, which drives a worm rear-axle through a propeller-shaft. The control is by a speed regulator of the multiple-contact type, operated by varying the resistance in the shunt field of the dynamo, and by shunting the series field of the electric motor.

Storage-battery buses are used in Chicago for carrying passengers between the railroad depots. In another city the street railway is now trying out a battery vehicle in short-haul feeder service. This bus is fitted with pneumatic tires and makes about 12 m.p.h. in ordinary operation. The limited speed and battery capacity have worked against the battery-driven vehicles, although they undoubtedly have a field for short runs on good pavements.

The trolley bus shown in Fig. 6, a road vehicle using the overhead system of the electric railway, is making headway on this side of the Atlantic, as shown by recent installations in Toronto, Canada; Minneapolis, and Los Angeles, Cal. The system on Staten Island, New York City, is to be extended by the use of 15 trolley buses now ordered for the New York City Department of Plant and Structures. Some of these will be used on a City Island line, in the Borough of the Bronx, New York City.

Many of the trolley buses look like rail cars fitted with rubber tires. However, there has been a tendency of late to follow closely gasoline vehicle design, thus using standard parts and making it possible to change from one type to the other by installing a gasoline engine and transmission apparatus.

A front-drive design has been worked out for the buses used in Leeds, England. The electric motors are under the front, and can be removed for repairs. It is claimed that this construction improves adhesion and tractive effort, reduces the consumption of electric energy and makes possible a 14-in. floor level.

The steam-propelled bus is not in commercial operation at present, although several experimental vehicles are being developed for such service. The Clarkson steam buses, which were popular in London because of their speed, quietness and ease of control, stopped operation in 1919, mainly, it is said, because of the increase in fuel cost.

CHASSIS COMPONENTS FOR BUS SERVICE

The principal parts of the bus chassis savor strongly of motor-truck practice, and bus individuality is gained by such features as more powerful engines, longer springs, battery and lighting generator and the like. Four-cylinder engines, with L-heads, are used on the

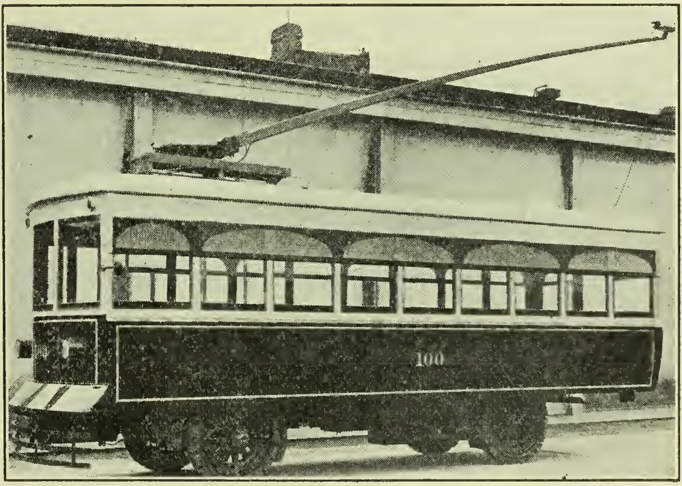

FIG. 6—A BUS OF THE SO-CALLED TRACKLESS TROLLEY TYPE

majority of buses. A few designs for city service have sleeve-valve units. Some I-head types are being supplied, mainly for the fast inter-city work. Many operators are talking in favor of six cylinders, because of the smoothness and flexibility they assure.

The data given in Table 1 indicate that engine speeds of from 1400 to 1800 r.p.m. are common, at least for city buses. These speeds need not be maintained for long periods, but they are essential, as is also quick acceleration, for threading city traffic, passing other vehicles on the open road and maintaining schedules when stops are frequent. Quiet operation at slow speeds is essential also.

Governors are not common on vehicles designed for bus operation. A driver to whom the lives of passengers can be entrusted will not require a governor to safeguard the vehicle.

In many localities the law requires that the gasoline tank be filled from outside the bus. Tanks therefore are being placed at the rear of the frame, or at the side between the wheels. Another arrangement is shown in Fig. 3, where the tank is placed under the floor of the body, and filled through a port-hole in the side of the body.

Ignition is mainly by magneto, this being favored by many operators because a magneto is easy to replace. With the growing practice of installing a battery and a lighting generator, battery ignition is likely to be given more consideration.

Thermosyphon and pump circulation are both used on buses. The latter system is used in the majority of the designs listed in Table 1. Radiators with replaceable cores are being developed for bus service. They illustrate the tendency to provide parts that will decrease the time required for adjustments or repairs.

The exhaust system is usually fitted with a valve connection so that the body-builder can install a heating system. The end of the exhaust pipe, and this also applies to heater outlets, should be placed at the extreme rear so that the gases will be discharged back of the body and not underneath it.

A smoothly operating clutch is an essential on buses. Jars and jerks not only increase maintenance costs, but they mean also continual discomfort to passengers. The service is so severe that this unit should be of liberal size.

Bus transmissions as a rule are of the four-speed sliding-gear type. They should be designed to stand up under a service involving 10 or more stops per mile and upward of 50,000 miles of operation in a year. The strain is intensified if the driver jumps to second or third speed when starting.

The maximum speed should be just high enough to maintain schedules from a transportation point of view, with a fair margin allowed to make up lost time. Passenger comfort requires steps between gears that are as nearly equal as possible. The low-gear speed must have power enough also to pull the vehicle through loose dirt, mud or snow.

Universal-joint design is especially important in the long-wheelbase bus. Bearing supports as well as extra joints are used to overcome the whipping effect of the long propeller-shafts. The angularity of these shafts is comparatively slight, so that full advantage can be taken of the shock-absorbing properties of the fabric joint.

The rear axle for the city buses is being made wider, of 72 and 74-in. gage, than on heavy motor vehicles. The underslung type of worm-gear axle is being used to a considerable extent to gain low floor-levels, while front axles with a center dropped sharply down between the pads and the yokes are being provided for city service.

Frame construction and proportions have been influenced appreciably by the development of the city-type bus. The law limits the overall width, so the bus body has had to be lengthened to get capacity. The overhang beyond the rear axle and in the back of the frame is limited also by considerations of safety and good design. All this has led to the long wheelbase, which seems to be longer with every new design. Frame sections have had to be made deeper on account of the great distance between the wheels. The frames have been widened at the same time to care for the 7 and 8-ft. bodies they must support. The demand for low floors has again been taken out on the frame, which is made with a kick-up over the rear axle so that the central part can be kept close to the ground. On one new design, shown in Fig. 7, this kick-up is found at both the front and the rear axles.

Owing to the severe service and the need for the utmost safety, the steering-gear and the brakes should be heavier even than for truck designs. Easy steering is important, to save the driver's strength.

A propeller-shaft brake is used on a number of bus models, undoubtedly because of its simplicity and the ease of securing equal braking on the rear wheels. On the Pacific coast four-wheel hydraulic brakes have been tried, for reasons of safety and to increase the life of the rear-wheel tires. The four-wheel air-brakes now being developed for heavy motor vehicles should be useful for large buses also.

Riding comfort is a bus requirement that has received close attention in the new designs. While underslung springs are installed principally to lower the bus floor, they undoubtedly improve the riding qualities of the vehicle. The change in wheelbase length that occurs on one side when one wheel strikes an obstruction is much less with the underslung construction, because the center of the rear axle is almost in line horizontally with the front eye of the spring, instead of being several inches below it, as with the conventional overhung type of construction.

Most of the bus designs have an auxiliary suspension that comes into play with heavy loads or severe road-shocks. The simplest is the so-called progressive spring, with extra leaves mounted below the main spring but at a flatter camber. Much the same effect is secured on the London buses by a volute spring attached to a bracket on the frame; or, the ends of the main spring are carried in sheaves, rubber blocks or small springs mounted inside the hangers. While these devices are used particularly on the rear springs, the front axle also is being better sprung, not only to protect the engine, but to carry the heavier loads thrown forward by the long wheelbase. Another method is to use the shock-absorbers developed for private passenger-cars. The pneumatic type, for example, is fitted to some of the California stages.

Pneumatic tires are almost universal on the small buses, but on the heavier types the large single pneumatic tires have the disadvantage of throwing the body high up into the air. This is being overcome by the application of dual tires on the rear wheels, which permits the use of six tires of the same size. The development of the "doughnut" design will help here, provided the brake diameters can be decreased to correspond.

Many operators in the cities are using solid or cushion tires, mounted on a resilient rim or cushion wheel. This combination seems to work out satisfactorily on fairly good pavements, although it has proved too heavy for the rear-axle mechanism on certain buses working over rough roads.

BUS BODIES

The old horse-drawn stages undoubtedly furnished the inspiration for modern bus bodies, and later technique seemed to be based also on the bodies built for motor trucks. At any rate, many of the first bodies were heavier than necessary for either economy or comfort. The timbers were too large, the ironing was too heavy and too much of it was used. Then, cheaper bodies were demanded; the pendulum swung too far the other way; bus bodies loosened up and became noisy; they began to fall to pieces and repelled instead of attracting passengers.

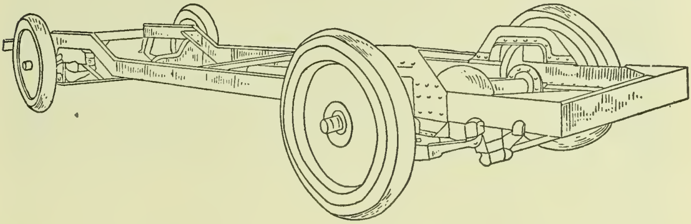

FIG. 7—BUS CHASSIS HAVING A KICK-UP AT EACH END RESULTING IN A FLOOR LEVEL THAT IS ONLY 19 IN. ABOVE THE GROUND

At first, quantity production of bus bodies was difficult because chassis mounting-dimensions were not standardized. Frame-widths and lengths, the location and the dimensions of rear wheels, the position of drivers' seats and controls and the height of the frame from the ground all seem to be different on every chassis. Bus bodies are now being turned out in quantities, however, partly because the sills can be altered to fit different chassis, and partly because of the great increase in the demand. The price and the weight have been decreased greatly, and, at the same time, there have been many improvements in detail construction.

The mounting on the chassis usually is designed to keep the floor as near the ground as possible. Many bodies now are set directly on the frame, so that the floor is only its own thickness higher. Longitudinal sills are placed between the frame side-members and supported between them. The part of the body projecting beyond the frame may be carried on cross-straps that extend the full width and are supported at the ends by pressed or built-up brackets riveted to the side-members; or, the support may consist of channels carried across under the main frame and bent so as to carry the outside edges of the body. These are riveted to the side-members by structural shapes.

In general, bus bodies are built up on a hard-wood framing, of ash or oak. Dies for the all-metal body would be too expensive, even should such construction stand the weaving and twisting to which semi-flexible chassis-frames subject it.

Panel material may be wood, sheet-metal or composition. Sheet-metal is applied usually as a sheathing, as is also sheet-aluminum. A wood veneer, protected on the outside by sheet-steel, is sometimes used. This material, it is claimed, has the resiliency of wood, and the easily finished surface of sheet-metal. This veneer, or plywood, is also made up as roofs, or used as a ceiling to furnish a smooth surface over roofs of tongue-and-groove board.

Climatic conditions and local preferences both seem to determine the type of body used in any given section. The open body is popular in England, where it is fitted to so-called motor coaches or char-a-bancs. The seats are unbroken across the body and are reached by individual doors. As a rule, the roof can be folded back. In this Country such a vehicle is favored in the South and on the Pacific coast; also, it is used in large cities and in the National parks where sight-seeing is the reason for the

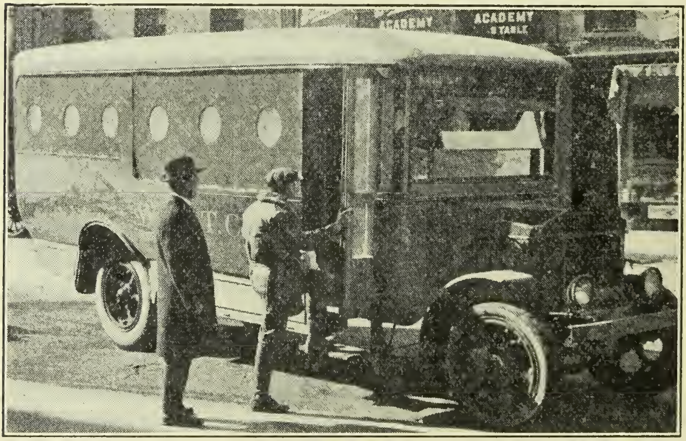

Fig. 8—Convertible Body Used for All-Year Service, the Roof Posts Being Enclosed by Double Curtains in the Winter

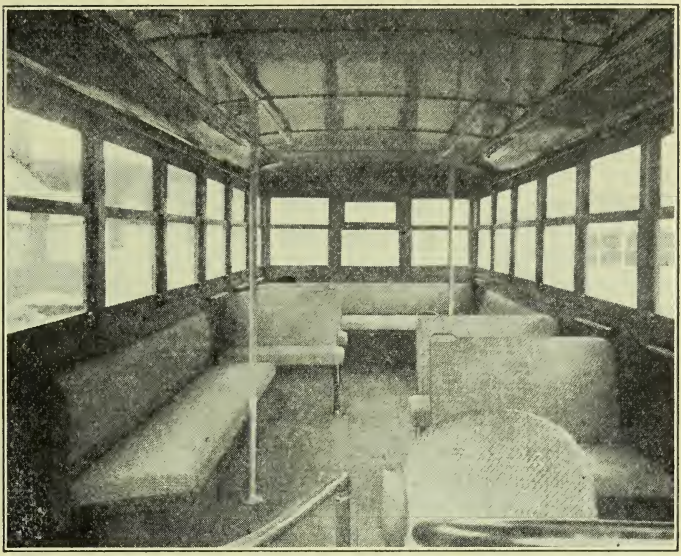

FIG. 9—THE SEATING ARRANGEMENT OF A BUS FOR CITY SERVICE WITH VERTICAL AND HORIZONTAL BARS TO TAKE CARE OF THE STANDEES

presence of buses. Sight-seeing buses work mostly in warm weather, but in some parts of the Country, as in New York State, an open convertible body, such as is shown in Fig. 8, is used the year round. This has a permanent roof supported on pillars that are built into the sides of the seats. The open sides are covered by double curtains during the winter.

The closed type of body is found generally through the North, East and Central parts of the Country. This body, with two longitudinal seats, is popular with operators because of its wide central aisle, large standee capacity and good ability to load and unload passengers. The longitudinal type is not so comfortable on long runs of 30 min. or more, where the passengers settle down and are not boarding or alighting at frequent intervals. In country service and for large single-deckers in city or inter-city work, it is being replaced by an arrangement such as is shown in Fig. 9, in which both cross and longitudinal seats are used.

COMFORT AND CONVENIENCE FACTORS

The success of most transportation systems depends upon their ability to attract a group of riders who travel not of necessity but because the service given is attractive and its use is a pleasure. This profit-making group is attracted by good-looking buses with pleasing lines, and also by steps, doors and seats that are comfortable and convenient.

The first "convenience" factor is on the outside of the body. The color of the paint has transportation value. Where buses are numerous, each route or line can use a distinctive color, so that patrons can recognize it easily. The electric railways operating buses are naturally painting them the same color as their rail equipment. In Bridgeport, Conn., all of the buses are painted the same color, but the belt rail is painted a distinctive color on each line. This stripe running around the outside identifies each route clearly.

To secure high visability, the buses of the Toronto Transportation Commission are painted a sagamore red. It is believed that this color induces both employes and passengers to keep the paint on the vehicles in better condition, and that accidents are reduced because the operators of other vehicles exercise greater care.

The would-be passenger, having picked out the right bus, is now ready to enter; but first he must allow passengers to alight, since it is not practicable to provide door space for two conflicting streams of travel. If he is to enter comfortably the actual step-height will not be more than 15 in., although from 4 to 6 in. can be gained by curb-loading. Tests on trolley-cars have proved that 15 in. should be the maximum step-height. Three steps, with one 16-in. and two 14-in. risers, showed an average loading and unloading time of 2.50 sec. per passenger. This was cut to 1.75 sec. per passenger, however, when the change was made to one 15-in. and three 10-in. risers, each of the four being slightly wider than the three used in the first test.

The width of steps should be at least 12 in., although this often is limited because of the necessity of keeping within the body lines, and clear of the chassis frame. The steps should be shod with safety tread, for the good of the passengers and to prevent wear. In current designs, the steps are of permanent construction, the outside folding type can be used but is not particularly satisfactory as regards durability or safety. Two steps only are needed with the low-frame buses, and these can be arranged to fit in the body very nicely.

The simplest form of door, assuming that the entrance is covered at all, is the collapsible gate used in Southern California. But more substantial construction is required in many parts of the Country. Several types of door are available. The sliding door frees the full width of the entrance, but makes necessary an expensive opening in the body. Two-leaf folding or jack-knife doors are therefore in more general use. These fold inward as a rule, as the outward folding type is frowned upon by the law-makers. Hinges at the front permit carrying the opening mechanism across to a handle that can be operated by the driver. The opening of the door may automatically turn on the step light at night.

The service door seems to be one of the most difficult parts of the body to keep in good working order. The trouble is usually in the door itself, although the manually operating mechanism may get out of order at times. On many buses the driver spends a good part of his time, often when he should be watching the road ahead of the moving vehicle, in trying to force a sticky door open or closed.

Doors should be at least 30 in. wide, although in practice they vary from 22 to 35 in. This can be compared with the exit-entrance of the one-man type of trolley-car, which is about 52 in. wide; but, as was said before, space is too valuable in the bus to provide a door that will carry two streams of passengers.

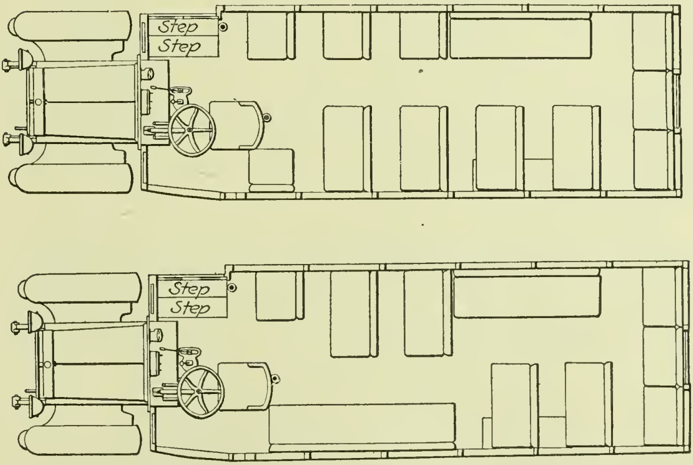

Fig. 10—Seating Layout of a 20-Passenger Bus (Above) for Inter-City Service and (Below) for City Use

Seats in the bus should be arranged in accordance with traffic conditions rather than the design of the vehicle chassis. The latter sometimes has to control, however, as in the placing of longitudinal seats over the wheel-housings. The result is a well at the rear of the vehicle, which must be reached through the neck of the bottle formed by the central aisle. This well is sometimes needed, but it is often used when the space should be filled with seats. Fig. 10 shows seating layouts for city and suburban service.

The standard cross-seat is 15 in. wide and 32 or 34 in. long. The central aisle is from 14 to 24 in. wide. The 34-in. seat is the better, since the narrower seat will not in reality allow the use of wider aisles. At the same time it may be uncomfortable for two persons. The real aisle-clearance is between the hips and shoulders of the seated passengers. With the 15-in. width, seat centers should be at least 28½ in., which gives only about 11-in. knee-room. These seat centers should coincide with side-post centers.

Seat construction has settled down to the spring type for cushions; springs are sometimes, although rarely, used in the backs. A patented seat, which is said to have given satisfaction on a 30-passenger bus, is supported on two pedestals, each of which is a pneumatic cylinder or dashpot. This really amounts to a shock-absorber directly applied to each seat.

Seat covers are of rattan ordinarily, with plush or leather for de luxe designs. Rattan is low in first cost and can be kept clean more easily. It is somewhat slippery, but this can be taken care of by the seat design; by raising the cushion at the front and recessing the back slightly. This gives more knee-room and comfort, and tends to hold the passengers firmly in place, even over rough going.

Floor coverings of carpet or linoleum are found on some vehicles, but for wear and hard service the floor is slatted, in the aisles and between the seats. The slats can be attached to flat strips of steel, from which they can be removed when worn. This scheme, which has been used in England, is said to make cleaning easier also.

The driver's comfort is equally as important as passenger comfort, and is taken good care of in the new designs. The seat is of bucket shape with good springs, and sometimes has an adjustment for height. A railing is placed so that passengers cannot crowd against the driver, wind and sun-shields are attached on the outside and a curtain or bulkhead prevents light from the rear being reflected on the wind-shield.

HEATING, LIGHTING AND VENTILATION

A heating system such as is shown in Fig. 11, using the exhaust gases from the engine, is installed on most buses put out within the last year or so. Previous to that, however, many buses carried small stoves during the winter months, which of course was unsafe and unsanitary. The first exhaust systems were makeshifts, with the gases led through pipes inside the body. While this was often effective, or at least provided satisfactory heating, it was a clumsy arrangement, hard to control and difficult to keep gas-tight. It was used, however, because operators claimed they could not get heaters of sufficient capacity, at least for large buses.

The problem seems to have been solved by the use of two heaters, placed close to the front of the body. The

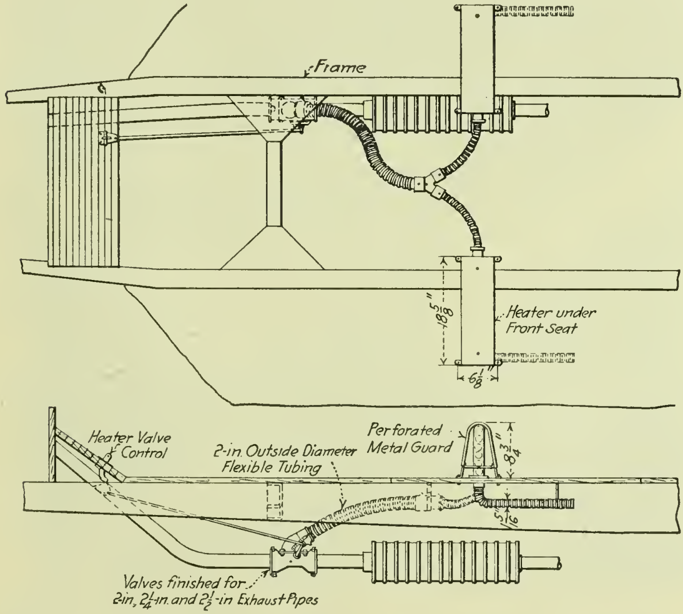
Frame
18 5/8"
Heater under Front Seat
6 1/8"
Heater Valve Control
2-in. Outside Diameter Flexible Tubing
Perforated Metal Guard
8 3/4"
5/16"
Valves finished for 2-in, 2 1/4-in. and 2 1/2-in Exhaust Pipes

FIG. 11—INSTALLATION OF HEATERS IN A 25-PASSENGER BUS SHOWING CONNECTIONS OF FLEXIBLE TUBING AND VALVE CONTROL OPERATED BY THE DRIVER'S FOOT

inlets for these join in a single connection to the exhaust pipe, through a valve controlled by a lever inserted in the floor-boards near the driver. Because of the compact arrangement, no provision is made for insulating the tubing leading to the heaters.

Good interior lighting is essential where passengers have the evening newspaper habit, although in country service, where the buses are used at night only on Saturday, it may not be necessary. Good lighting is needed also for displaying advertising cards, which promise to be a considerable source of income to the bus operator.

The 2-cp. bulbs used in so many buses may help in collecting fares, but are of little service to passengers. It is recognized now that larger generating and battery capacity must be installed. The 300-watt generator system for the Fifth Avenue Coach Co. buses, although in some respects a special job, indicates that the larger equipment is available. The 12-volt battery system seems to have found a place, although the 6-volt design is used widely.

The present tendency is to mount dome reflectors on account of the low head-room. Prismatic and opal reflectors have been designed however for bus service. These require practically no more head-room space and are said to be much more efficient than the domes. The large buses are being fitted with six to eight bulbs, of from 21 to 32 cp.

Identification at night is a requirement peculiar to the bus. Many designs are fitted with a bulls-eye, of special color, at the top of the roof in front. This permits the would-be passenger to pick out the bus when it is some distance away, and the bulls-eye lights up the destination sign when the bus comes closer. Purple and green running-lights serve the same purpose.

Bus ventilation, in its simplest form, depends upon the service door for the admission of fresh air, and upon vents at the sides or the rear to exhaust polluted air. Most doors admit a slight amount of air even when closed, unless they overlap at the top and bottom, when they are practically draftproof. Air is admitted also through openings across the front of the body, above the driver.

The latest practice is to insert ventilators along the center of the roof, with a screen or grating in the ceiling. These ventilators may be automatic in operation, not requiring any adjustment; rain cannot enter but the air exit is always ready for service. Windows are being equipped with adjustable curtains and sash, safety catches, rubber weather-strips, anti-rattlers and upper stationary sash with Florentine glass. Wire grating or wood slats are placed outside to keep each passenger wholly inside the bus.

Pockets for windows are of two types. The overhead pocket requires good head-room, and heavy top construction with a higher center of gravity. Then there is the side-pocket type that takes up valuable space needed for seats and aisles. The overhead opening seems to be favored, undoubtedly because the windows are easier to handle.

FARE-COLLECTING DEVICES

The collection of fares, while more of a traffic than an equipment matter, is giving much trouble to bus operators. The necessity for simplicity and accuracy has led

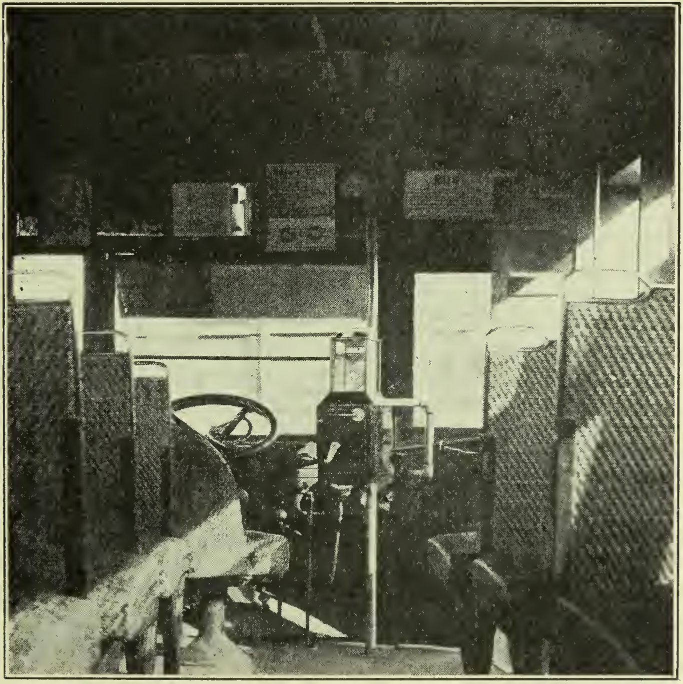

FIG. 12—REGISTERING TYPE OF FARE-BOX MOUNTED ON A STANCHION AT THE RIGHT OF THE DRIVER

to the application of mechanical devices, such as illustrated in Fig. 12, to save work for the driver, insure his honesty and speed-up service.

A money-changer, fastened to the side of the bus or at some other convenient place, is the first step; next some type of fare-box, with perhaps an overhead register that the driver pulls to record fares. The latest arrangement is what may be described as an instantaneous registering lamp-post type of fare-collecting device. This records each fare as deposited, and is driven by a flexible shaft connected to one of the accessories on the bus engine. The use of such a mechanism should make unnecessary the regulations that forbid the driver to collect fares while the vehicle is in motion.

State and Local Regulations

Laws and ordinances by the hundred, almost by the thousand, have been passed to "supervise and regulate" the bus. The safety and comfort of the traveling public are protected by a host of these regulations, many of which are more honored in the breach than in the observance. In common with other users of motor vehicles, the bus operator is limited as to weight, dimensions and speeds. In addition, he often must pay special taxes that influence the type of equipment.

To assure safe operation, presumably, the carrying capacity of the bus is limited. In some States, in New Hampshire, for example, the manufacturer's rating is taken as the legal limit, and no standees are permitted. New Jersey allows 50 per cent of the seating capacity to be carried as standees. Connecticut allows two more than the number of seats provided. The District of Columbia permits one standing passenger for each 1.5 sq. ft. of floor area, this being calculated on the area remaining after deducting the space occupied by seats and 10-in. knee-room for longitudinal seats. In Maryland the maximum carrying-capacity is the total length of seats divided by 16, except when the result exceeds the carrying capacity. This method, in effect, bars out standees.

An emergency door that can be opened from the inside in case of accident is required in a number of localities. Other equipment insisted upon, for safety reasons, includes speedometers, inside lights, extra tires, skid chains, warning device, and a fire protection device.

The requirements for heating, lighting and ventilating are usually indefinite; when enforced they depend upon the interpretation of the authorities. Closed buses in the District of Columbia must have a minimum temperature of 40 deg. fahr. between November 1 and April 1. Interior lights, of 2 cp. or more, are demanded in Ohio and other States. In Maryland each bus must have two ventilators.

In many States the taxes are based upon seating capacity, and the larger vehicles pay more per seat than the smaller. Florida exacts from the buses used as common carriers a fee of $5 per seat for 7-passenger or less capacity, $7.50 per seat for 7 to 17-passenger capacity, and $10 per seat for more than 17-passenger capacity; in addition there is a tax of 75 cents per 100 lb. of gross weight of the vehicle and the load. Connecticut and Maryland also have systems whereby the fee shoots skyward for the larger buses.

The regulatory authorities may even supervise the purchase of equipment, approve drawings and specifications and compel the improvement of those already in service. It is too early yet to judge what the result of this supervision will be. At present it seems to make for better buses. With wise regulation, and with improved highways that will permit lighter vehicles and more economical operation, the bus, already with such a wonderful development, will have an even more wonderful future as an instrument of transportation not only on the streets of our cities but also in inter-city service.

THE DISCUSSION

R. E. Plimpton:—What I have tried to do in this paper is to bring out the transportation viewpoint, the viewpoint of the operator who is operating mainly the smaller lines. What I call here "the country bus" is really the small vehicle that often is operated over rough roads or in service where the traffic is light.

The conclusion of the paper refers to regulation very briefly, only as it affects the design of the vehicle, but that thing is coming more and more. President Bachman referred to the competitive feature or possibilities of the bus, but I do not see any competition there at all. Mr. Green has referred to the bus as a public utility. I think that our institutions are firmly grounded on the regulation of public utilities and that the bus is coming more and more into that class. Already, more than half the States in this Country have some form of state regulation and, while they started in with restricting the lines and the fares charged, they will continue that and will get more into equipment, not only for safety, but for the comfort of the passengers; and, as that goes on, I think engineers and builders ought to keep more closely in touch with the regulatory authorities, so that they really will be informed. At present, the State authorities are taking many steps connected with design that they are sorry for when they realize what the steps mean. They require certain types of body, or certain types of tire, perhaps, and then, after they pass the law or the regulation they find out that it will not work, and that is not a good thing for anybody.

To show what some of the smaller operators are up against when they go into the business and also to show the viewpoint of a small city, I will read an extract from a Newburgh, N. Y., newspaper. They have about 50 buses now in Newburgh. Outside of New York City, I suppose it is one of the largest centers of bus operation in New York State, in spite of the fact that it is a comparatively small city. The city lies on the Hudson, so that one side of the territory is cut off; but, in spite of this, they have all this bus operation. This newspaper says:

> Any one contemplating going into the bus-transportation business will do well to figure the ultimate cost before going just far enough to have to turn back with a loss on his hands. First, there is a cost of one bus, which may be anywhere from $1,000 to $10,000. There is required a liability bond of $10,000, which costs $592. In addition, there is a New York State license fee, which costs $67.50. There is also a Federal license fee of $20, and $225 for property insurance on the bus. Thus it will be seen that it will cost, exclusive of the price of the bus, $904.50 before a fare has been collected, to embark in the bus business. The bus-transportation business is no plaything for men who have any regard for their money. The day when a man could run a tumble-down car in and out of Newburgh and call it a bus because no one cared enough about what he called it to put up a kick is probably gone forever. A man involved in the bus business in this section is regarded as a transportation unit and that line of industry is evidently not to be meddled with by those not knowing what they are about. If you can afford it, there is money in the bus-transportation line. If you cannot afford it, there is not.

Chairman C. F. Scott, of the Meetings Committee, told me some time ago that he took it as a matter of fact that practically every city and town in this Country had some form of bus transportation. I am inclined to amend that by saying that practically every city and town has had experience with bus operation, which is a different thing, because the dead bus lines mount up into considerable numbers. Of course, that applies in the city also. I think the reason for that is not altogether due, although it is partly due, to inadequate design of the vehicles; it is due also to a lack of transportation knowledge on the part of the operators. It is only a matter of time before they will acquire that knowledge as a result of these failures.

MERRILL C. HORINE:—Does not the second item in Table 1 of Mr. Plimpton's paper, the approximate entrance-height in inches, really mean the floor height above the ground? The Mack engine bore is 4¼ in., and the front axle is of I-beam construction. If he means by entrance height the height of the floor at the entrance, that will make it clearer; but by entrance height I should say is meant the first step, which of course cannot be as high as shown.

G. A. GREEN:—Referring to Mr. Plimpton's remarks in regard to the possible future employment of six-cylinder bus engines and six-wheel buses, our theory is to concentrate on the reduction rather than on the addition of parts. As to the six-cylinder engine, for normal bus operation in the average American city, we believe a four-cylinder engine of about 300-cu. in. capacity is adequate; and assuming proper design, speed and load variations can be taken care of by carbureter adjustments or gear-ratio modifications. If this assumption is correct, then certainly it is folly to add two more cylinders, thus increasing first cost, weight, number of parts and fuel bills. Of course, these remarks apply without regard to whether the engine design is poppet or sleeve valve. But it should be pointed out that the much desired silence and smoothness of operation are comparatively easily achieved with four-cylinder engines of the sleeve-valve type.

In regard to the six-wheel bus, here again there are additional parts and, in all probability, additional weight and cost. In our judgment the value of this construction has not been proved. In short, it would seem a more logical plan to concentrate our energy on the improvement of the four-wheel system that has given and is giving an exceedingly good account of itself, rather than expend any great amount of time and money on the development of a theory that in our opinion has not yet emerged from its early experimental stages.

In Mr. Plimpton's paper, a suggestion is made in regard to painting buses different colors for different routes. From our viewpoint this would be unwise, for with a public utility there should be the fewest possible restrictions placed on vehicles. The underlying thought is that any vehicle should be capable of operation by any driver on any route at any time.

George W. Cravens:—I think this whole matter comes back to the principle stated by Mr. Green; first, know your business, and then design something suitable to your purpose.

I am interested in several companies, among which is one that is experimenting with and will bring out a steam bus. There is no doubt that there is much to be said in favor of steam from the standpoint of acceleration and smooth operation, but I will not take time for that. However, in this bus-transportation problem you have a similar problem to that of the street-railway man in the last analysis. You must take care of your traffic at the least cost per passenger-mile.

In our investigation of motorbus requirements it seemed to us that standardization of body sizes and wheelbase boils down to approximately 25, 40 and 55-passenger as being good standard sizes. How nearly does that agree with Mr. Green's and Mr. Plimpton's determinations?

Secondly, I believe the time has come when it is necessary to clarify the bus situation. That has been done in the passenger-car and to a certain extent in the truck and tractor fields. It is time for a company to go into the business of building a vehicle from the ground up that will be consistent as well as suitable for its purposes. The time of makeshifts and of trying to redesign something you have, to fit a new condition you know nothing about, has passed. I believe that eventually some company similar to the Fifth Avenue Coach Co. will go into the business of constructing buses for sale that will be built along standard lines, and you will take them or go without them, the same as is now necessary when we buy a truck or a passenger car.

Mr. Green:—We have analyzed the seating-capacity situation as nearly as we can, and as a result of our analysis we are building vehicles carrying, with the single-deck bus, from 25 to 29 passengers, and with the double deck, from 50 to 55 passengers. Undoubtedly there are fields for other vehicles, but we think the largest field is for buses of these capacities. Of course, there is also a field for the so-called speed wagon, seating around 15 passengers, but with vehicles of this character, the driver's wages consume a large proportion of the revenue, so the natural tendency will be to increase the seating capacity gradually to make up for this.

HORACE L. HOWELL:—With reference to the question as to the proper seating capacity of a bus, I should consider that it would depend chiefly upon the character and class of service; that is, if it were a high-speed proposition, the single-deck high-speed type of bus would be the answer, with a seating capacity similar to that of the Fifth Avenue Coach Co., 25 or 29. Should the bus be required for urban transport service at slow speeds, a double-deck structure would seem more proper if the overhead clearances would allow it. In this case the number of riders would govern the seating capacity.

The first successful type of bus used by the London General Omnibus Co., London, England, was known as the B-type bus. It had a seating capacity of 34 passengers. That bus was used for a number of years and approximately 1000 of them were commandeered by the British Government for transport service in France during the recent war. A number of them are in operation today; but, owing to the growth of the riding habit and the inadequate capacity of the fleet, the construction program of the London General Omnibus Co. called for a larger vehicle with a greater seating capacity. The company therefore established a program for its future operations, to meet the increased riding habit, by constructing the K-type bus having a seating capacity of 46 passengers and, later, the S-type bus having a seating capacity of 54 passengers. The operating statistics of the company show that the cost of operation for the larger passenger-carrying vehicle is not much more than it was with the B-type bus and it may be expected that, on account of the added seating-capacity and because of the comparatively small difference in operating costs, the consequent operations will be much more successful from a financial standpoint.

CORNELIUS T. MYERS: — In reference to capacity, it would seem that wherever bus lines have become established and their patronage grows, the larger seating-capacities can be employed profitably. In a great many districts very likely we find that the smaller bus will be the first installation, not only on account of first cost but because the regular service cannot be given as traffic has not been developed.

MR. PLIMPTON:—One of the most encouraging things in connection with the whole bus situation is that there are several companies now specializing in the construction of vehicles to be used as buses. We must bear in mind that although the traffic is light there is a great volume of other work done by the operator of the country bus. He very often carries light packages and freight; he may deliver milk or newspapers or do errands for people; and he usually has some other business. But for the territory he serves he performs a real service. In Italy and France this has been recognized and Government subsidies have been applied to this "country service," just to keep the buses going.

In connection with Mr. Horine's query, what I had in mind was not the step-height, but the approximate floor-height; that varies somewhat, but that is what the figures were intended to cover.

Mr. Cravens suggests three sizes of bus for standardization, the smallest being of 25-passenger capacity. Undoubtedly, there is a considerable field for a smaller vehicle of say 16 to 18-passenger capacity, and it should be given due consideration.

NOTES ON MOTOR TRUCKS[1]

By Cornelius T Myers[2]

After pointing out that the publication of articles in the trade and technical journals, to the effect that very considerable weight-reductions in motor-truck construction with consequent savings in gasoline and tires are possible, works an injustice to the motor-truck industry and is misleading, the author outlines some of the reasons why such weight-reductions are very difficult to effect, as well as the possibilities of standardizing axle details. The use of aluminum to effect weight-reduction is commented upon and the various advantages claimed for metal wheels are mentioned. In the latter connection the author points out that, while these claims may be true, they are unsupported by reliable data.

The greater part of the paper is devoted to an account of a series of tests conducted by a large coal company to determine the relative merits of wood and metal wheels on its trucks. Four trucks, each equipped with wood and metal wheels on diagonal corners so as to secure, as far as possible, an equalization of conditions were employed. The tests lasted over a year, and at their conclusion it was found that the average wear of the tires mounted on metal wheels was about 13 per cent greater than that of tires mounted on wood wheels.

The question of unsprung weight is discussed, as is the importance of reducing chassis and body weights to a minimum, but it is pointed out that a reduction in these weights does not necessarily mean a resultant saving in the gasoline consumption or the tire expense. Lubrication of the various parts of a motor-truck chassis also receives attention, the annual cost of truck repairs due to poor lubrication of the chassis being given as from $15,000,000 to $20,000,000, in addition to which there is a loss of earnings while the trucks are being repaired of from $30,000,000 to $35,000,000. The superiority of oil over grease as a chassis lubricant is emphasized.

In conclusion it is pointed out that the next few years will see bitter competition in motor-truck service and that refinement of detail and simplification of operating features will be emphasized. The need for the unexperienced buyer to secure expert advice regarding the

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[1] Pennsylvania Section paper.

[2] M.S.A.E.—Consulting engineer, Rahway, N. J.

> design of the vehicles offered him is stressed, it being stated that this practice would clarify the situation for both the buyer and the seller.

Articles keep appearing from time to time in various publications to the effect that our motor trucks are far too heavy and that they can be lightened materially if we but use aluminum and alloy steels in this or that form; also that these weight-reductions will effect big savings in gasoline and tires. These articles prophesy a Ford passenger-car descendent in a motor truck, and are not at all just to the motor-truck industry in that they lead purchasers to believe that the near future holds out the fulfillment of such a promise. There are also statements in regard to unsprung weight, quantity production, wheel construction and similar subjects that are likely to be misleading to those not in close touch with the subject. Doubtless, in time, great advances in construction will be achieved, but this will be rather by a large number of short steps than by a revolutionary change in design.

We often hear the alloy-steel man talk of much lighter frame-sections, etc., but we have reached a point today in motor-truck construction where the possibilities of chassis weight-reduction are approaching a safe limit, regardless of application of higher-grade steels, because the limiting features in many parts of the chassis design are those of deflection and of metal thickness sufficient to hold rivets and avoid buckling, rather than tensile stress. One magazine article states, "In frame construction the old heavy charcoal-iron has been almost entirely superseded by the pressed steel which, with about half the weight of the former, possesses nearly double its strength." This would result in making each pound of steel in the frame about four times more effective. But this is not the case. Assuming that "charcoal iron" refers to the rolled standard structural section, it is worthy of note that several very widely known and highly rated truck builders still use frame steel in this form, and also that their frame weights are, if anything, heavier today than 8 or 10 years ago for similar load capacities. Further, the weight of frames made of pressed steel for similar load capacities by well-established competing truck builders averages about the same as the weight of frames made from rolled sections, as is brought out in Fig. 1.

The same principle applies to axles. Lighter front-axles can be made of high-tensile heat-treated steel, but under working loads the deflections would be such as to seriously misalign the wheel spindles and steering connections. Also, such an axle, when accidentally bent could be straightened only with difficulty and would have doubtful value thereafter as a load carrier. Rear axles, too, could be lightened in the same way, but at the expense of misalignment under load deflections that would be fatal to their driving mechanisms.

POSSIBILITIES IN REAR-AXLE STANDARDIZATION

A comprehensive paper on Rear Axles for Motor Trucks was presented last month before the Metropolitan Section. A casual glance through the paper by a member of the Society's Standards Committee will cause

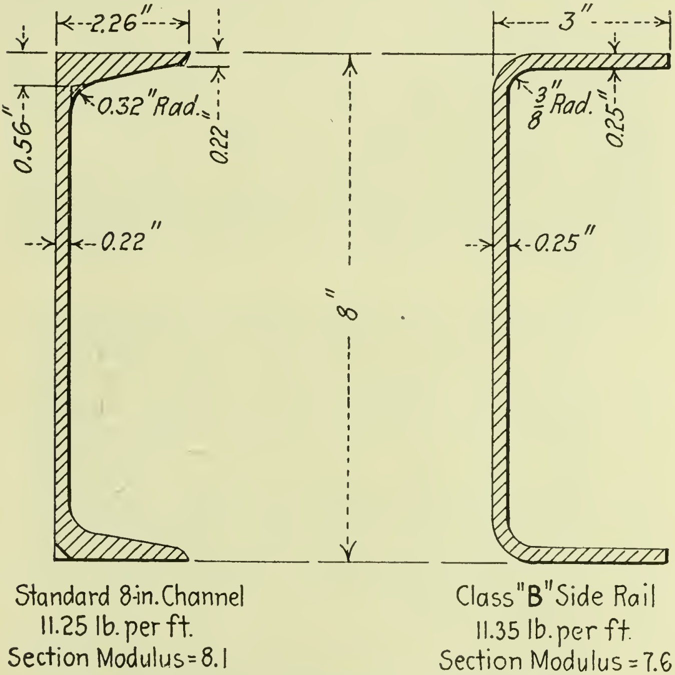
2.26"
0.32" Rad.
0.56"
0.22"
0.22"
8"
3"
3/8" Rad.
0.25"
0.25"
Standard 8-in. Channel
11.25 lb. per ft.
Section Modulus = 8.1
Class "B" Side Rail
11.35 lb. per ft.
Section Modulus = 7.6

FIG. 1—ROLLED AND PRESSED STEEL FRAME SECTIONS

The Standard Rolled Channel Is 1 Per Cent Lighter and 7 Per Cent Stiffer Than the Pressed Channel. The Latter Permits the Use of Large and Easily Made Fillets in the Frame Bracket, Although the Rolled Section Is Often Beveled Off to Give the Same Advantage. Bolts Through the Flanges of the Rolled Channel Require Special Wedge-Shaped Washers to Hold the Heads True him to contemplate with awe the enormous investment in axle spare parts it must entail to service the different makes of motor truck already in operation. Further contemplation increases this by the amount of different manufacturing tool equipment necessary to produce them, and the terrific expense it must entail to keep the many little armies of salesmen each fighting their twenty-sided battles over unimportant details. Eighty million dollars is spent every year to sell trucks, a large part of it going to support fruitless detail arguments. The truck industry now generally recognizes that it must sell transportation and not details, and there seems to be a great opportunity to standardize on axle details. Everyone at interest will benefit by this, for all the well-known types of rear axle are firmly established in the trade and give very good results in use. The user is not interested in axles if they carry his goods without disturbing his peace of mind. When accidents happen or wear takes place, however, he wants repairs made quickly and cheaply. This not only saves him money but keeps him in the frame of mind to spend money in buying another truck. I will put before you certain features of the various axle designs that seem to offer more or less attractive fields in which to pursue the well-defined and enormously profitable policy of standardization that has been steadfastly fostered by the Society of Automotive Engineers.

(1) For each of seven truck sizes, standard spring center-lines and spring widths can be established; giving a reasonable variation, for the time being, by using spring-pads that are wider than necessary. The spring centers will depend on frame widths, a subject that is already being attacked with every reason to expect success. These established, the spring clips and nuts will fall readily in line. Even the spring-pad drilling and spring-seat height offer possibilities of being reconciled between chain-drive and internal-gear axles, and also among single-reduction, double-reduction and worm-drive axles

(2) Standard brake-drum diameters, widths and metal thicknesses are now being discussed in the Standards Committee and tentative dimensions have been suggested. Once these are determined, the door will be open to the crying need for standardizing the dimensions of brake-linings, rivets, hinge pins, cams or toggles, diameters of camshafts and bush-

ings, fits for camshaft levers, clearances and movements, release springs and all the minor details that are so annoying when replacements are needed at an inopportune time or in an out-of-the-way place. There will be double paths to follow here by those who differ as to having more than one brake inside the drum, and also by the adherents of the external brake

(3) Once the brake-drum width is selected, the wheel spoke is located. The wood wheel can fall in line readily here and the steel wheel also. Our tire sizes are standard and also their fits on the wheels. Standard practice with reference to front-wheel hubs has already been adopted. The rear hubs also should yield to the same analysis and attack, though the solution will not be so simple. The hub details depend upon the type, size and location of the bearings used

(4) The wheel bearings cause a considerable diversification. Either ball or roller bearings can be employed in each of the five types of axle most used. But of these five types single-reduction, double-reduction and worm-drive can be grouped in both the fixed-hub or full-floating designs; while chain-drive and internal-gear drive also can be grouped. Thus, for each size truck axle 16 variations in bearing sizes, types, fits and spacing may be reduced to six. Possibly a thorough study of the situation would reduce these still further, as was found possible in the front-hub bearing standardization, in which over 200 hub variations were boiled down to 10, which are served by only 14 bearings

(5) Fits for hubs, flanges, hub-caps and wheels can be standardized correspondingly, once the bearings are fixed

(6) The fits at the outer ends of spindles and drive-shafts can be standardized in accordance with the classification given under (4); and if the wheel tracks are held close the spindles and drive-shafts themselves offer possibilities. Items 4, 5 and 6 will react on the spring-pads mentioned under (1), and help reconcile some discrepancies that at first may occur there

(7) Differentials can be standardized. The subject is now under consideration. A reasonable number of sizes to cover the range of torques to be transmitted, is all that is necessary. It should make little difference in what type of axle they are used.

There would be a series for ball bearings and a series for roller bearings.

(8) The above accomplished, the fits for the inner ends of drive-shafts would fall in line

(9) Bevel, spur and worm gears all offer great opportunities for standardization. This would lead to a similar accomplishment for the bearings and bearing fits on worm spindles and bevel-pinion spindles, but these can be standardized whether or not the gearing is standardized

(10) Adjusting devices for differential bearings and spindle bearings also offer a field

(11) The spindle fits for universal-joint flanges have already been standardized

(12) Clevises and pins for brake-lever connections have been standardized. Possibly the levers themselves are susceptible to this work

Here, then, are 12 featurees of rear-axle standardization, some of which have been solved. All types of rear axle have been so perfected and perform so well that a standardization of all assembly dimensions and common details is the next big move in the progress of the industry.

Alloy-steel crankshafts are negligibly lighter for the same stiffness than those made of less expensive carbon-steel. The alloy-steel will resist wear somewhat better than carbon-steel if the latter is not case-hardened. However, wear can best be reduced by improved lubrication.

Weight-reduction by means of aluminum is not new in engine crankcases, gearboxes, radiators and other smaller parts. Temporarily, cast iron is being used to a large extent, because of the high cost of aluminum. Aluminum has been used successfully in some parts of passenger-car rear-axles. Some experimental truck-axles have been made of it. But this metal has a structure too coarsely crystalline to make it readily serviceable under the heavy and uncushioned shock loads a truck rear-axle must endure. Some alloy of aluminum may solve the problem, however.

A few truck wheels have been made of aluminum but with unsatisfactory results. The most promising truck wheel today is the wood-spoke type but of lighter construction and better detail than those made in the past. There is much misunderstanding of the functions performed by motor-vehicle wheels, and much misinformation has been circulated in the last few years. The average man's idea of a wheel covers a device that revolves on an axle spindle and carries a tire. The common opinion is that the stronger a wheel is, the better. This accounts to a considerable extent for the development of various types of metal wheels, cast steel, malleable cast iron, rolled or pressed disc and even forged steel, that are offered and championed mainly on two contentions that (*a*) they are stronger than the wood-spoked wheel; and (*b*) they are more accurately made and more true than the wood-spoked wheel. Both of these contentions are true. It is claimed, therefore, that

(1) Tires mounted on them will last longer
(2) Gasoline consumption will be less
(3) The truck will be less liable to be put out of commission due to accidental damage to the wheels

It is claimed also that cast-metal wheels weigh less than the conventional wood wheel and therefore decrease the unsprung weight, 1 lb. of unsprung weight being variously estimated in these claims as equivalent in load effect to 5 to 10 lb. of weight above the springs.

WOOD AND METAL WHEELS

For several years there has been considerable controversy as to the relative merits of motor-truck wheels made of wood in the usual manner and those made of cast steel or malleable iron. The following are typical statements, taken from manufacturers' circulars or from the advertising pages of trade papers with reference to cast-metal wheels:

> The permanent true roundness of .... wheels makes for lower fuel bills, and increases the mileage obtained from tires. Users tell us of increases of from 10 to 30 per cent
>
> They increase tire life; they decrease fuel consumption
>
> To move a pound of weight in the wheels requires as great an expenditure of tires, power and fuel as is needed to move 10 lb. above the springs..........
>
> .... wheels are as much as 100 lb. lighter per set than other types of wheel
>
> Their trueness and roundness add 10 to 25 per cent to the mileage obtainable from tires
>
> They reduce unsprung weight and lower tire costs

Repeated inquiry among those who made these statements convinced me that they were made on a basis of supposition and hope, and not on established data. Possibly the claims were true, but no data supporting them were forthcoming. Neither was there any sound basis for the statements made in regard to unsprung weight, or the inferences drawn therefrom.

While it is true that the manufacturing limits for roundness of wood wheels are several times as great as those for steel wheels, the variations allowed the wood-wheel manufacturer are so small as to be negligible in the operation of the truck. The inequalities of the road surface are so much greater and the cushioning value of the tire is so large that the effect of the relatively minute wheel variations cannot be measured in any operating expense. The greater accuracy of the metal wheel seems, therefore, to be but gilt on the lily.

In checking up wheel weights it was found that in commercial production the cast wheels were in most cases heavier than wood wheels built for the same load capacity, especially in sizes up to those suitable for 3½-ton trucks. In sizes for 5 to 7½-ton trucks wheel weights in some instances more nearly coincided but the wood wheels still averaged somewhat less in weight. In comparing weights the weight of the hub, flange and brake-drum or sprocket assemblies were included in the weight of the wheels. It did not seem likely that the heavier metal wheels would give greater tire mileage; but our absolute knowledge of the effects of varying amounts and proportions of sprung and unsprung weight is still very limited. In spite of the fact that the superior tire-mileage claims were based on *lighter weights* for metal wheels, it seemed that some comparative data should be secured, rather than leave the truck user to conjecture on a point so important in the operation of his vehicles. The motor-truck user has to spend his good money for everything he gets, and it is important for him to know whether the extra price he has to pay for cast wheels really comes back to him in tire savings. A practical test under every-day conditions, but with proper safeguards, is necessary to secure this information.

TESTS MADE WITH FOUR TRUCKS

A company that trucked large tonnages of coal in one of our big cities, was interested in this subject, equipped four of its trucks and conducted such a trial. It operates a large fleet of trucks, mostly of 6½ and 7-ton capacity. It keeps a careful record system that includes mileage, tonnage and operating costs. Its trucks are well kept-up. The drivers are well trained. The operation of the trucks is very capably supervised. Altogether the conditions were as near ideal as could be expected.

On trucks of 5-ton capacity and up, wood wheels are thought to be at more of a disadvantage than on trucks of smaller capacity, and 6½ and 7-ton trucks were therefore used in the test. The trucks carried full loads on each trip and operated on regular schedules and predetermined routes. Half of the trucks were equipped with wood wheels on the right-hand end of the front axle and the left-hand end of the rear axle; the wheels at the other diagonal corners of the truck being cast-metal as is shown in Fig. 2. All the wheels were equipped with new tires, of not only the same make, but made at about the same time with the same compound and cure. All tires contained approximately the same thickness of rubber. The rated loads on the tires were within commercial limits of good practice per inch of width, and these loads did not vary widely for the different trucks used.

Two chain-drive trucks, Nos. 162 and 168, and two worm-drive trucks, Nos. 172 and 175, were selected for the test. The wheel equipment was as set forth in Table 1.

Each wheel was carefully stamped for identification. The metal wheels for trucks Nos. 162 and 168 and the wood wheels for trucks Nos. 172 and 175 were new, but the wood wheels, for trucks Nos. 162 and 168 and the metal wheels for trucks Nos. 172 and 175 had had con-

TABLE 1—WHEEL AND TIRE EQUIPMENT

Truck	Front		Back	
No.	Wood	Metal	Wood	Metal
162	*Left*	*Right*	*Right*	*Left*
	Interlock spoke	Hollow spoke	Interlock spoke	Hollow spoke
	36 x 6 S	36 x 6 S	40 x 6 D	40 x 6 D
168	*Right*	*Left*	*Left*	*Right*
	Mitred spoke	Hollow spoke	Interlock spoke	Hollow spoke
	36 x 6 S	36 x 6 S	40 x 6 D	40 x 6 D
172	*Left*	*Right*	*Right*	*Left*
	Mitred spoke	Cast disc	Mitred spoke	Cast disc
	36 x 6 S	36 x 6 S	40 x 7 D	40 x 7 D
175	*Right*	*Left*	*Left*	*Right*
	Mitred spoke	Cast disc	Mitred spoke	Cast disc
	36 x 6 S	36 x 6 S	40 x 7 D	40 x 7 D

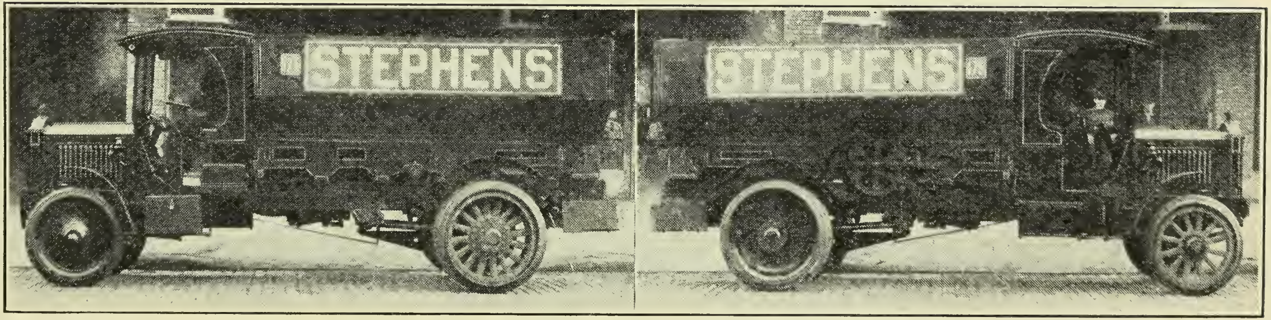
STEPHENS
STEPHENS

FIG. 2—VIEWS OF OPPOSITE SIDES OF ONE OF THE TRUCKS TESTED

siderable service on these trucks. This arrangement, it will be noted, placed the wood wheels and also the metal wheels at diagonal corners of each truck, but alternated them as to rights and lefts on the two trucks of the same make. This assured that the average wear would take place under the same conditions for both wood and metal wheels. The superintendent said that his records showed that in general the tires on the right-hand side of their trucks wore-out somewhat more rapidly than those on the left-hand side. This is probably due to the slight slope of the road toward the right, which loads the tires on the right-hand side somewhat more than those on the left-hand side.

The front wheels on all four trucks carried 36 x 6-in. single tires. The back wheels on trucks Nos. 162 and 168 carried 40 x 6-in. dual tires, while the back wheels on trucks Nos. 172 and 175 carried 40 x 7-in. dual tires; 24

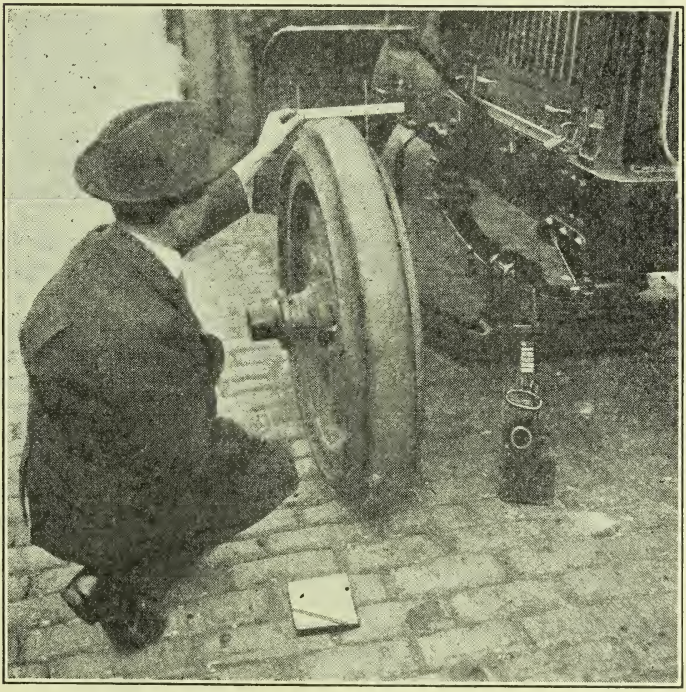

Fig. 3—Taking Measurements of the Height of the Rubber Above the Tire Band

TABLE 2—STARTING DATES OF TESTS

Truck No.	
162	Nov. 11, 1920
168	Nov. 12, 1920
172	Nov. 19, 1920
175	Nov. 18, 1920

tires in all being included in the test so as to get a fair average.

Before purchasing the tires the matter of furnishing a uniform quality was taken up in detail with the tire factory, and after careful inquiry tires were selected that had all been produced at about the same time, and that the factory stated could be depended upon to be practically uniform and give comparable results in this test.

Each steel tire-band was stamped for identification so that damage to the serial number on the tire would not cause confusion. Each tire-band was then marked at six places, equidistant around the rim. At each of these places the height of the rubber above the flange of the band was measured carefully with a gage made especially for that purpose as shown in Fig. 3. The average of these measurements was taken as the initial height of the rubber in the tire. All measurements were checked by two persons. Tires selected were applied at random on the wheels, and it was impossible to distinguish any difference in them.

As each truck was equipped with new tires, records of mileage and tonnage were carefully compiled. The starting dates of the tests are given in Table 2.

In addition to the regular operating records of these trucks (they were operated entirely in accordance with the business demands and not given any special routes or service) the tires were given a special inspection about once a month. This was to keep track of any unusual wear, bad cuts, base separation, etc.

Due to the mild winter of 1920-1921 these coal trucks did not cover a very great mileage for the first month or

TABLE 3—CAPACITY AND LOAD PER INCH OF TIRE WIDTH

Truck No.	Pay-Load Capacity, tons	Average Weight per Inch of Tire Width, lb.
162	6½	720
168	6½	743
172	7	702
175	7	696

so. Each load carried, however, was a capacity load, as can be seen from Table 3.

On March 4, 1921, inspection of the tires showed but little wear, and all of them were in good condition. Measurements were taken. The rates of wear in inches per 1000 miles and the mileage at that date are shown on charts reproduced in Figs. 4 and 5. The average rates of wear show that for the front wheels the tires on the wood wheels were doing about 2.75 per cent better than those on the steel wheels. On the rear wheels the tires on the steel wheels averaged 0.5 per cent better than those on the wood wheels.

In considering the results given herein, it must be borne in mind that the differences in rate of wear between individual trucks is not directly comparable, because they all operated under somewhat different conditions. For instance,

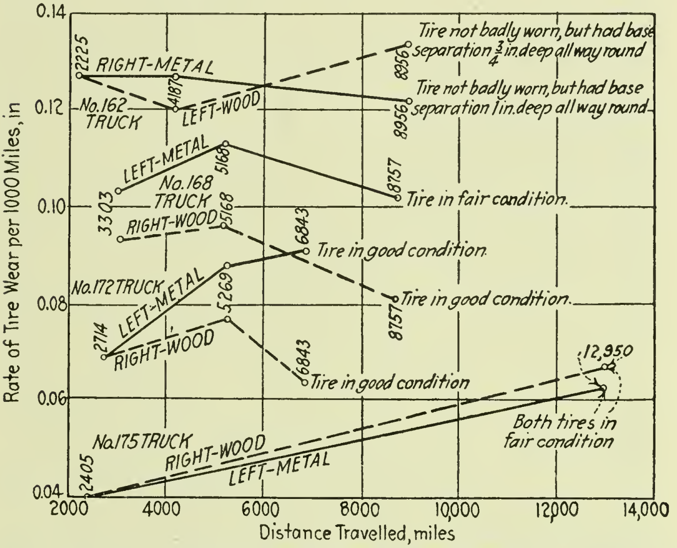

FIG. 4—RELATION BETWEEN THE RATE OF TIRE WEAR PER 1000 MILES OF TRAVEL AND THE DISTANCE COVERED FOR FRONT WHEELS

(1) Their routes and mileage varied

(2) Their tare-weights were different, as is indicated by the weights per inch of tire width given in Table 3

(3) They had different drivers; often an important item

(4) The size of the rear tires was different
(5) The load per inch of tire width varied
(6) They operated at different average speeds, though at the same maximum speed
(7) The springs of the chain-drive trucks were not the same as those on the worm-drive trucks

To compare, therefore, the results in tire wear on trucks Nos. 162 and 168 with those of trucks Nos. 172 and 175 is misleading on account of the variables. True comparisons are possible only where all essential conditions are the same. It was realized that this difficulty would confront any attempts to get the comparisons in tire service on wood and on cast metal wheels; hence the use of four different trucks and the alternate diagonal arrangement of the wheels. Here the average results are an indication of the relative service, because both front tires and both sets of rear tires on each truck were subjected to the same operating conditions, with the exception that possibly the tires on one side of each truck might have slightly more load to carry than those on the other side. To counter this exception the "hand" of the wheels was alternated on similar trucks; wood wheels being placed on the right-hand end of the front axle on one chain-drive and one worm-drive truck, but on the left-hand end of the front axle for the other chain-drive and worm-drive. This arrangement was followed also on the rear axles, as will be seen by referring to Table 1.

During the next 10 or 11 weeks the trucks were in active service, and on May 20 another series of tire measurements was taken. The inspection showed that all the tires seemed to be wearing uniformly. There were no large cuts, no hangnails, and no base-separations. Even from a casual glance, however, the tires on some of the metal wheels showed more wear than those on the wood wheels on the same truck. Figs. 4 and 5 give the individual rates of wear and mileages up to May 20.

Through the late summer and early fall there was comparatively little for these trucks to do, and it was not until some of the rear tires showed signs of disintegration that the next measurements were taken, on Dec. 1, 1921, a little over a year from the time the tires were new and first measured. These measurements showed that for the eight front tires the average wear of those mounted on metal wheels was 9 per cent greater than the wear on the tires mounted on wood wheels; also that for the 16 rear tires the wear of those mounted on metal

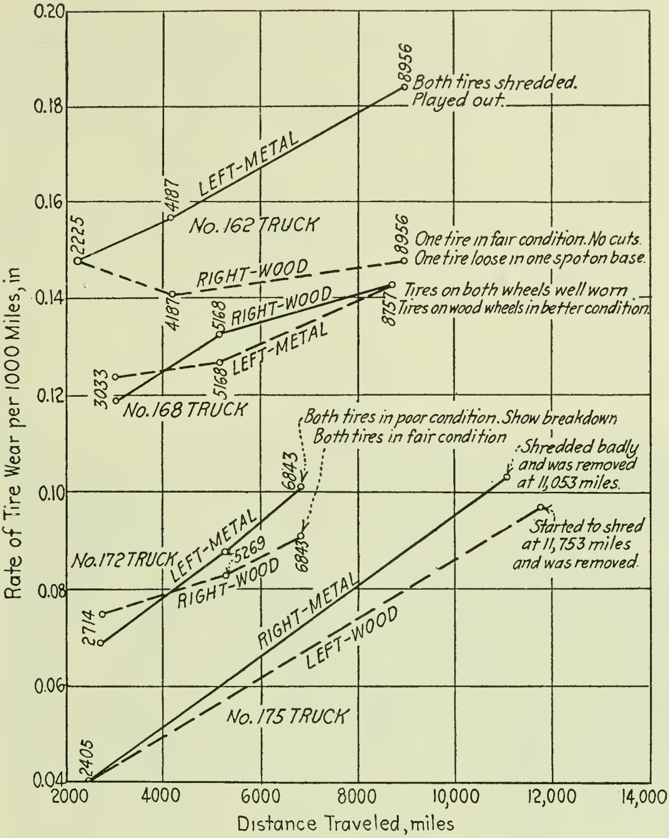

Fig. 5—Relation Between the Rate of Tire Wear per 1000 Miles of Travel and the Distance Covered for Rear Wheels

wheels was nearly 11 per cent greater than the wear of the tires mounted on wood wheels. These percentages are based on the average rates of wear per 1000 miles over the entire period. The charts show the cumulative results at the end of each period when measurements were taken.

Of the tires on rear metal wheels none exceeded the tires on the rear wood wheels in mileage. On truck No. 168 they were equal, but on the other three trucks the tires on the wood wheels gave superior results. Not only were the tires worn more on the metal wheels but they showed signs of disintegration to a much greater extent than those on the wood wheels. Late in the fall of 1921 the superintendent in charge of the operation of trucks, who had previously thought the metal wheels were the more desirable, stated that there was no use taking any more measurements, for anybody could see that the tires on the wood wheels were giving better service.

On the front wheels none of the tires had worn-out, but those on truck No. 162 showed that base separation had begun. On the rear wheels all the tires were well worn; some of them were shredded and about to be replaced, as can be seen from Figs. 6 and 7.

As the tires wore down and their cushioning effect diminished, it was noticeable that small failure cracks appeared sooner and developed into larger cracks in al-

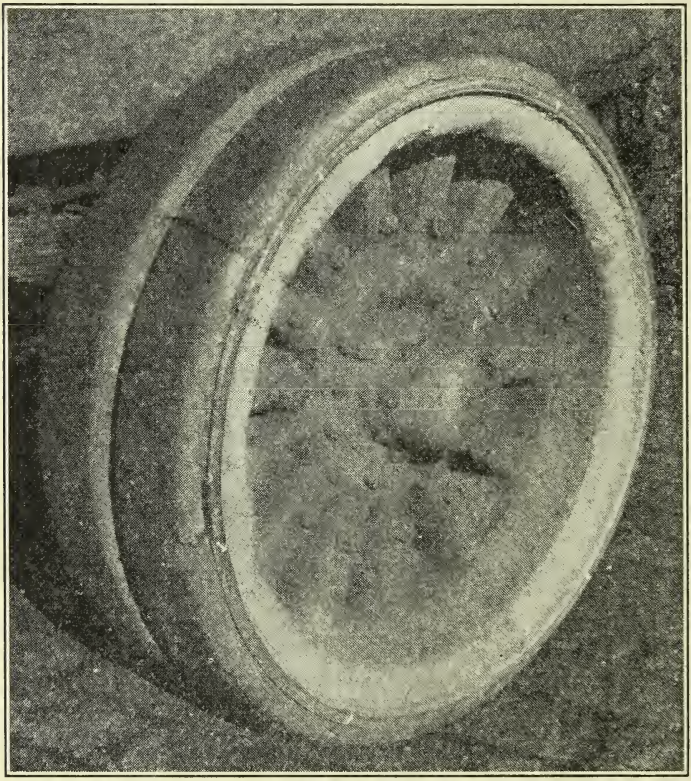

FIG. 6—REAR WOOD WHEEL ON THE SAME AXLE AS THE WHEEL IN FIG. 7 AND ITS TIRE AT THE CONCLUSION OF THE TEST

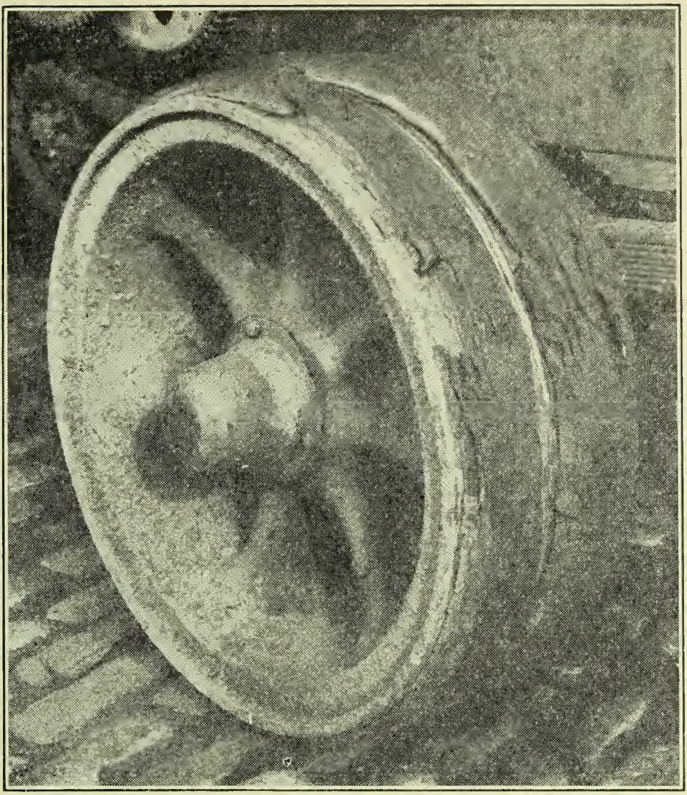

FIG. 7—REAR METAL WHEEL ON THE SAME AXLE AS THE WHEEL IN FIG. 6 AND ITS TIRE AT THE CONCLUSION OF THE TEST

most every case in the tires on the metal wheels. As the steel wheels in every case exceeded the wood wheels in weight it might be thought that this was due to the greater inertia of the heavier wheels. This may have been true to some extent, and it appears reasonable but, if that were the case, how can the differences in the tire mileage between the chain-drive trucks and the worm-drive trucks be explained? The worm-drive trucks gave better results and the unsprung weight of their axles considerably exceeded that of the chain-drive. Neglecting wheels and tires, the weight of the worm-drive rear-axle parts was about 1400 lb. greater than that of the chain-drive axle. These results need careful interpretation as will be indicated later.

UNSPRUNG WEIGHT

It is my opinion that the relative effect of a difference in unsprung weight is modified to a considerable extent by the presence of the springs above the axle and, of course, the speed at which the truck operates, as well as by the thickness and hardness of the tire itself. In other words, against vertical impact there is the cushioning effect of the springs, and to some extent of the frame and load itself, as well as of the tire. The tire therefore does not have to withstand the full force of the road impact when delivered in a vertical plane. When the blow is delivered in a horizontal plane, however, the springs are not able to cushion the blow and the tire must absorb the full impact if the wheel is rigid. Nor is the tire designed to withstand sidewise or skid loads as efficiently as vertical loads.

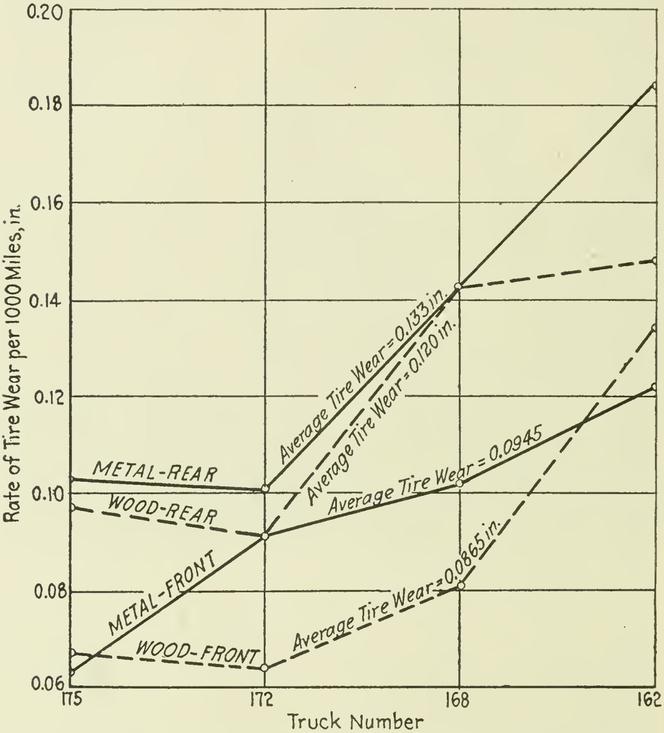

FIG. 8—RELATIVE RATES OF WEAR OF FRONT AND REAR TIRES MOUNTED ON METAL AND WOOD WHEELS PER 1000 MILES

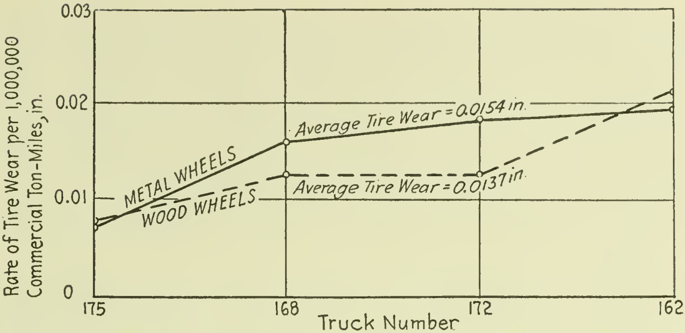

FIG. 9—RATE OF TIRE WEAR PER 1,000,000 COMMERCIAL TON-MILES FOR FRONT TIRES

One of the important functions of a wheel is the ability to act to some extent as a cushion to the savage side-thrusts of rut and curb. The wood wheel is several times as resilient as the metal wheel and affords this protection, while the metal wheel does not flex to any appreciable extent. Thus with wood wheels the tire has some assistance from the resilient spoke at the time it needs it most. In the extreme case of a heavy side-skid against a curb, the wood wheel breaks, acting like the fuse in an overloaded electric circuit, and protecting more expensive parts.

TIRE WEAR PER THOUSAND MILES

Figs. 4 and 5 show the results of the test in terms of mileage and rate of wear per 1000 miles. As the results for the different trucks vary so much, it may be thought that the test was not a fair one because the individual

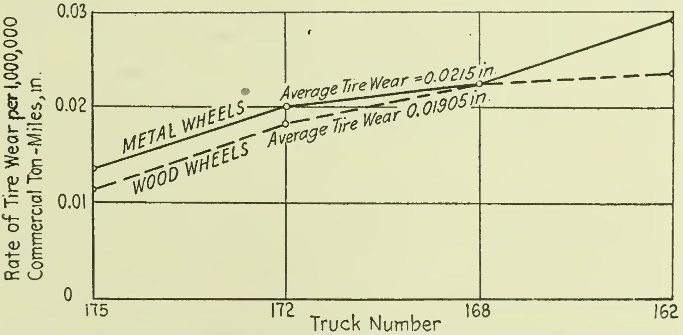

FIG. 10—RATE OF TIRE WEAR PER 1,000,000 COMMERCIAL TON-MILES FOR REAR TIRES

tires themselves must have varied in composition. In this connection it should be noted that the rates of wear of the tires of each truck form a distinct group. The differences in the rates of wear, therefore, are not due to differences in tires, but to differences in trucks, their drivers or operating conditions. Figs. 4 and 5 show this plainly. Fig. 8 shows the rates of wear per 1000 miles on the front and rear tires, plotted to show the relative difference between the tires on the wood and the metal wheels.

Figs. 9 and 10 show the rates of tire wear per 1,000,000 commercial ton-miles. Here we see that trucks Nos. 168 and 172 are not very far apart in tire wear when certain allowances are made. The front tires on truck No. 168 show 7 per cent less wear than those on No. 172. The rear tires on truck No. 172 show 15 per cent less wear than those on No. 168. It should be noted, however, that truck No. 172 had the advantage of 4 in. more in width of tire on the two rear wheels, which probably accounts for a great part of this difference. On the basis of commercial ton-miles we find that for the front wheels the rate of wear was about 12½ per cent greater for the tires mounted on metal wheels, while at the rear the rate of wear was about 13 per cent greater for the tires mounted on metal wheels.

Truck No. 175 was in charge of a very careful driver and in every way was kept in as nearly 100 per cent of first-class condition as was possible. During the last few months of the test this truck was hired out to some contractors, as not enough coal was being delivered to keep it in that service. On this contract work it covered greater distances than on the coal deliveries, and making extra trips it rolled up a big commercial ton-mileage. This is a case of exceptionally good performance by a carefully groomed truck with a very good driver.

Fig. 11 gives a final comparison between the rates of wear per 1,000,000 commercial ton-miles of tires on wood and on metal wheels. It averages the rates of wear of the six tires on each truck, or 24 tires in all. Twelve of these were on wood wheels and 12 on metal wheels.

In setting forth the results of this test it is not intended to convey the impression that solid tires on wood wheels will give uniformly from 10 to 15 per cent better performances than on metal wheels. The number of trucks tested was far too small to justify such a conclusion. The results are strongly indicative, however, and cast grave doubts on the unsupported statements of those who have sold us steel wheels. It seems safe to say that the claims for metal wheels, cited early in this report, are by no means justified. Those who make such claims should demonstrate them beyond doubt by concrete performance.

The logic of lighter and more resilient wheels would indicate a better performance by the wood wheels, even in the event of no great amount of confirmative data.

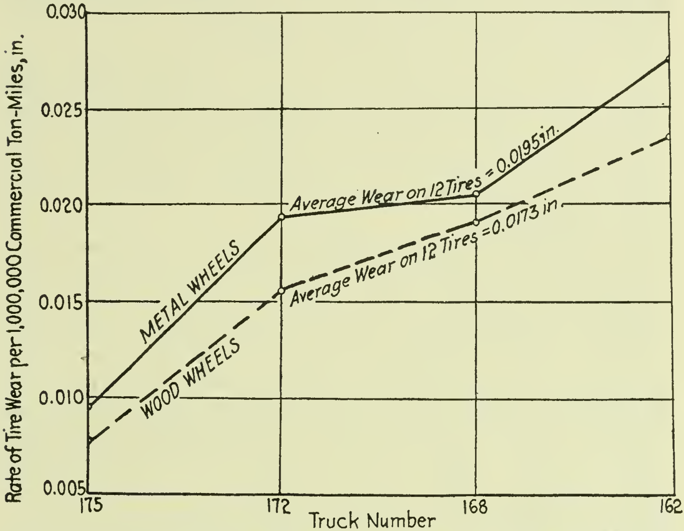

FIG. 11—COMPARISON BETWEEN THE RATES OF TIRE WEAR PER 1,000,000 COMMERCIAL TON-MILES OF TIRES ON METAL AND WOOD WHEELS

The claim for better performance by a metal wheel because it is more nearly round and true is really a point of minor importance, because the road inequalities so vastly exceed the very small variations that occur in a well-made wood wheel that even a worn tire will practically nullify them.

It has been claimed also on many occasions that solid tires wear less on metal wheels than on wood wheels, because the former "radiate" the heat generated in the tire at a more rapid rate than the latter. This is another theory that has not by any means been established. The test described above shows that the difference in the rate of wear was greatest in favor of the tires on the wood wheels during the heat of summer, directly contrary to the above theory, when the metal wheels should have had the greater relative cooling effect and saved the tires. The theory is defective in that it assumes that the heat due to the rolling of the tire causes more rapid wear, and that the radiating ability of the wheel is a large factor in cooling the tire. However, the air surrounding the rubber is probably of major importance in cooling the tire; and if the metal wheels are painted, as is almost universally the case, the paint film probably offsets the ability of the metal itself to transfer heat from the tire-band to the air around the spokes.

It is true that during 1917, 1918 and 1919 some very poor wood wheels were produced and assembled under trucks, and these have been held up as horrible examples. These were produced under extenuating conditions, and are by no means typical of the millions produced prior to that time and still giving good service if they have had reasonable care.

In regard to weights, the cast-metal wheel is seldom as light as the ordinary wood wheel, which is really heavier now than it need be. This item also touches on the much bandied question of the relative importance of unsprung weight to the total load carried. This is most often cited in discussions of different types of rear axle. It is reasonable to suppose that the lighter the axle the less will be the force of the blow transmitted through the springs to the frame when the wheels encounter bumps in the road surface. The relative effect of heavier blows by heavier axles is not readily discernible on the load carried, however, if the truck has well-designed springs, especially if these springs are well lubricated. Then, too, the heavier the axle the greater the proportion of the blow absorbed in the tire, due to the greater inertia of the axle. So, it would seem that this would be accompanied by a greater tire wear under the heavier axle. But again there are no reliable supporting data.

Of the four trucks used by the coal dealer in the tire-mileage test already mentioned, two had worm-drive rear-axles, the heaviest type, and two had chain-drive rear-axles, the lightest type. The rear-axle loads per inch of tire width were about 7 per cent greater for the chain drive than for the worm drive. Leaving out all reference to unsprung weights, one would expect the tires on the chain-drive trucks to show somewhat more wear, perhaps 10 per cent, per commercial ton-mile. The total weight of the worm-drive axle, wheels, tires, etc., was

70 per cent in excess of the weight of similar parts on the chain-drive axle. If 1 lb. of weight below the springs is equivalent to 5 lb. above the springs, then the tires on the worm-drive axle were loaded 31 per cent more per inch of width than those on the chain-drive axle. If the more fanciful ratio of 10 to 1 is used, the excess becomes 68 per cent. In any event, if the proponents of the unsprung weight theories are to be believed, the tires on the worm-drive axle should get the worst of it to a substantial degree, but, as a matter of fact, careful records show that the average rate of wear of the rear tires on the chain-drive trucks was 53 per cent in excess of those on the worm-drive trucks.

Certainly these results do not indicate that a light axle saves tires nor, however, do they prove the opposite, because the conditions of the test were made such as to afford comparative data on the wheel service, and were *not* such as to enable one to draw fair comparisons on an unsprung-weight basis. They are cited as an example of data that are being submitted repeatedly in advocating one type of design over another in making truck sales, and buyers should be on guard against such statistics and comparisons.

Nothing in this paper should be construed as deprecating the importance of reducing chassis and body weights to a minimum. For every idle pound of such weight removed, a pound of useful load can be added. Weight-reduction, however, does not always mean a proportionate gasoline or tire saving. Other factors enter into these items of expense, and those who generalize expose themselves to the same error as those who have allowed their imaginations to over-emphasize the importance of a minimum unsprung-weight. To date there are no data on which to make safe comparisons, and a full exposition of the subject may establish a different feeling from that which exists at present.

Engine weights can be reduced if improvements in fuel and lubrication allow us to use higher compressions and maintain oil-films on rubbing surfaces. The air-cooled engine will effect a considerable weight-reduction if perfected for motor-truck service, and this is not at all impossible.

For years the intensive study given to the main units of the chassis, engine, radiator, gearbox, axles, frame, springs, steering-gear, etc., has absorbed nearly all the efforts of designers. The details by which these units were assembled in the chassis, kept in proper relation, and controlled, have suffered by comparison. As a consequence they have been responsible for many breakdowns and a heavy upkeep expense. At present these details are receiving closer attention but there is still room for improvement.

Chassis Lubrication

An outstanding subject at the present time is that of chassis lubrication, the lubrication of the numerous small pins and bearings connecting the working units of the chassis. All mechanical motion, in the millions of forms that surround us on every hand, depends upon good lubrication for its continued existence. The lubrication of steering pivots and connections, universal-joints, spring-bolts, pedal-shafts, brake-shafts, radius-rod pins, clutch-release bearings, unit support pivots, etc., of motor trucks, in most cases has been cared for crudely in the chassis design. Consequently these parts have been subject to severe wear and entailed heavy upkeep expense. Lack of good lubrication in engines, gearboxes or rear axles is quickly attended by an ominous noise and dire disaster; hence the studious development for these units of lubricating devices and systems that are practically automatic in operation and require infrequent attention. Engines will go from 200 to 500 miles on a charge of oil. Gearboxes and axles will go 3000 to 5000 miles on a charge of lubricant.

When we turn to the consideration of the chassis bearings mentioned above we find most of them equipped with lubricating devices of the crudest sort, in spite of the fact that in many cases, on spring-bolts and universal-joints, the pressures per square inch are as high as those encountered on the crankshaft bearings of the engine. In addition, the ends of the bearings are, for the most part, exposed to the dust and mud of the road, while atmospheric moisture can enter almost at will. As a result these bearings deteriorate rapidly, entailing gradual loss of power, hard riding, difficulty of control, spring breakages, squeaks, rattles and a general looseness that adversely affects other parts of the chassis. The fact that the chassis bearings can function at all under these conditions, and neither produce a startling outcry nor actually breakdown for some time, has been responsible for much of the neglect of lubrication on the part of both the manufacturer and the user.

The user pays a big bill for this neglect, however, and the truck builder feels the reflection of it in sales resistance. In this Country the estimated direct cost of truck repairs due primarily and secondarily to poor chassis lubrication is between $15,000,000 and $20,000,000 per year, a truly staggering bill. On the other hand, at only 50c. per hr. the labor charge for lubricating properly our trucks with the devices now furnished would be over $40,000,000 per year. It is evident, therefore, that the truck user has actually been justified to some extent in his neglect of these parts, for, even if he gave them the attention that the builder desires, he would still have some repair bills, say $5,000,000 per year, and the total direct economic loss would exceed $25,000,000 annually.

The indirect loss to the user, the loss of earnings while the trucks are being repaired, runs from $30,000,000 to $35,000,000 annually. This is not always taken into consideration by owners; otherwise they would pay more systematic attention to chassis lubrication, and make a more insistent demand upon truck manufacturers for means to allay this heavy expense.

Except at one or two points, grease as a chassis lubricant has been acknowledged by close students to be inferior to oil and it is becoming less popular. Grease is a dirt collector and carrier; it is not uniform and is only part lubricant; it will not spread over a bearing by capillarity; it can be forced over the slack side of an oscillating bearing without reaching the load side, and as it exudes from the ends of the bearing it gives the misleading impression that the bearing is lubricated; it entails a high labor charge for application. Oil is a much better lubricant, for it carries no inert matter; even if dirty it can be filtered as it is fed; it spreads completely over a bearing by capillarity; it can be fed automatically; it can be led from one point to another in series; it is cheaper. The average oil-cup, grease-cup or fitting, however, is an excrescence, easily damaged or broken off. It admits dirt and moisture to the bearing, and its repeated filling calls for a heavy labor item.

An early attempt to embody thorough and automatic chassis lubrication was made in 1916 by a Pacific coast truck builder who embodied small oil-reservoirs, or magazines, in the brackets that held the chassis bearings. In this system the oil is filtered and fed by wicks to the bearings whenever the motion of the bearing calls for it.

About once a month the reservoirs are filled from a spout can. This magazine system was so satisfactory that the 20,000 Class B Trucks for the United States Army were so equipped. After 3½ years of hard service in the New York district 300 of these trucks were overhauled, and the inspection showed that less than ½ per cent of all the chassis bearings fed by the magazines showed appreciable wear. This is a vast improvement over the usual condition of such bearings, which in many cases must be renewed entirely, both pins and bushings, after 2 years' service. The surplus oil from the spring-bolts spread down the springs, kept them soft and flexible, and practically eliminated spring breakage. A magazine type of lubricator for spring-shackles that is equipped with a rapid fill opening is illustrated in Fig. 12.

By piping the various magazines to a few convenient points at the side of the chassis, they can be filled in less than 10 min. The first applications of this system had no provision for regulating the feed, which was much more than sufficient. Later improvements regulate this so that the magazines need not be filled oftener than once in 2 months. The national labor charge for truck chassis lubrication can be reduced in this manner to $500,000 per year. The chassis-bearing repair-bill can be cut to an insignificant figure, as can also the lay-up loss due to these repairs. A conservative estimate of the direct and indirect savings on our 1,010,700 commercial cars is $50,000,000 per year. This alone would buy 15,000 new chassis. It would buy 20,000 chassis if these were built in one plant each year.

The next few years will likely see much active competition in motor-truck service. Refinement of detail and simplification of operating features will be emphasized.

Many buyers today are not sure as to what features are essential and which ones are non-essential. Few buyers have any fundamental knowledge of motor-truck design or the compromises necessary to produce a successful design. In purchasing they stress this or that point without absolutely knowing its value or the effect of it on the cost and operation of the truck as a whole. This makes for differences of opinion among builders as to what the trade wants. Neither buyer nor seller gets any permanent benefit under these conditions.

Frequently the buyers' experience does not qualify them to pass judgment on the propositions that are offered. Once the full picture is before them, there are

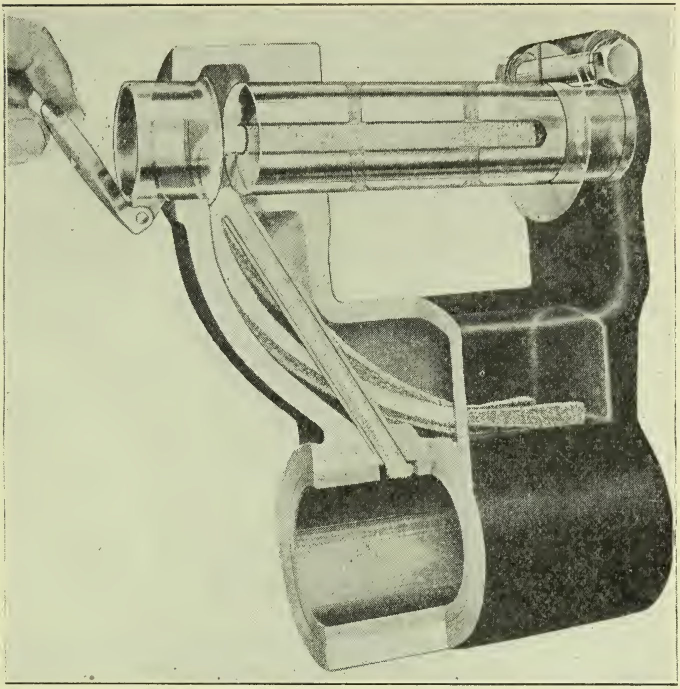

Fig. 12—A Magazine Lubricator for Spring-Shackles with a Rapid Fill Opening

few executives who are not qualified to judge the situation and make a proper decision. Such decisions would soon clarify the situation for both buyer and seller.

In the long run motor-truck design should tend toward such standardization as has been adopted to so great an economic advantage by the Master Car Builders' Association, which permits railroad rolling-stock to be repaired quickly and cheaply in any part of the Country. This need not restrict initiative or halt improvement, but will make any innovations conform to good practice, as determined by careful analysis of cause and effect, and long study in the application of the results.

THE DISCUSSION

President B. B. Bachman:—Regarding the brief analysis that Mr. Myers made as to the relative stiffness and strength and the uses of higher grade material, I am prepared to agree, although I think there is a possibility of misinterpretation of what he said. We all know that the technical points that he brings out regarding the form of a structure and its effect upon stiffness, and the lack of effect of the quality of material, are well taken; however, I feel that Mr. Myers has gone to extreme lengths to a large extent in this paper in endeavoring to clear up some possible misstatements that have been made in the endeavor to market certain classes of product. I believe that the advantages of elasticity have not been fully realized and, while I do not think the frame or the front axle are members in which the element of elasticity is a desirable factor, there are portions of the chassis structure in which elasticity can be obtained properly by the form of construction and design.

The Hotchkiss drive, in itself, permits weight reduction in the chassis design. The reduction of chassis weights and the advantages that can be obtained thereby are a result of study and an analysis of the power requirements and the engine size used in the vehicle, in conjunction with the gear-ratios, the gearbox and the rear-axle. Such considerations have a tendency to reduce the tare weight of the chassis, with a consequent realization of economy in operation.

Probably no one is more sympathetic to standardization than I am. I have had the honor of being Chairman of the Standards Committee for 4 years, and I have been associated with the standards work in one capacity or another for many years, practically since the inception of that work in the Society. But I do not concur fully with what Mr. Myers says with regard to the desirability of the standardization he outlined. If I interpret correctly the closing paragraphs of his paper, he is making a plea for interchangeability of essential components of motor vehicles that would undoubtedly be detrimental to initiative and the development of design.

He uses as an analogy the work accomplished in railroad circles by the Master Car Builders Association. In railroad practice a freight car may leave New York City with a load and, without having its load changed, cross the Continent, traversing an air-line distance of 3000 miles, and possibly going many hundreds or even thousands of miles more than that, depending upon the methods of routing. In that time it would pass over a number of different railroad systems. It is evident that there should be a considerable degree of interchangeability. The most obvious standard is that of the gage of the tracks. In the matter of repairs in the various roundhouses the stocks carried must be interchangeable. In spite of what we have seen in the last few years, under abnormal conditions, of long truck-transport, it does not seem to me that we shall ever approach a condition entirely analogous to that of the railroads. While the benefits of interchangeability that Mr. Myers points out have a certain foundation and can be defended along logical lines, I believe there is a reverse side of the picture.

There is room for reasonable differences of opinion on the matter, without any actual antagonism from either side. It is possible that, as the motor-truck industry gets out of its swaddling clothes, some of the things Mr. Myers has outlined will come to pass. Certainly, history tells us that the views of men of vision are needed to point out things that some of us who are more closely tied to details think cannot be accomplished.

In regard to the subject of wheels, Mr. Myers has reported the results of some tests with the idea, as I understand it, of refuting certain sales statements that have been made. The data he has brought forward would not sway me one way or the other. I feel that the results that he points to, and that he has definitely qualified, as secured from a small number of trucks, really do not tell us anything. At the same time the claims he is attempting to refute have not, so far as I know, been established otherwise than from a controversial standpoint.

On the other side of the case, Mr. Myers refers to the fact that during the last few years an unsatisfactory quality of wood wheel was manufactured and sold with disastrous results. That factor alone is, I believe, responsible for the rise of the metal wheel to the quantity production it has attained. The only fear I have had with regard to the wood wheel is from the standpoint of the felloe stock. I have had no fear of the spoke stock. The felloe stock, particularly in the case of large-size wheels, should be improved. Designs have been proposed, and some of them have had a considerable amount of practical trial. That appears to eliminate the difficulty of wooden felloes and offers the possibility of satisfactory service from substituting steel while retaining the essential characteristics of the wood wheel. I believe that the serviceability and satisfactoriness of the wood wheel and of the metal wheel are dependent more upon honest workmanship and good quality of material than upon anything else. Either construction will give good service under proper conditions.

The questions and theories that have been raised with regard to flexibility, radiation of heat and the like are all matters of conjecture and possibly matters for future research. So far as I know, none of us has had time to investigate and obtain data to substantiate any one of these various thoughts or theories.

C. T. Myers:—Our chief problem in truck design is to secure a construction that will offset impact blows. The more flexible and more resilient the construction is, the better it will resist impact blows. I agree with Mr. Bachman that the frame weights and all other weights should be cut down to the limit. The point I tried to make is that there are many people who do not know what they are talking about at all and never had to spend months and years in getting a truck on the road and keeping it there, but make claims of the possibility of vast weight-reductions that, so far as we can see at present, are not tenable. Some of them claim to reduce the weight of a motor truck by one-third. That is absurd. Accomplishing a 10 to 15-per cent weight-reduction in the average truck is doing exceedingly well and it takes extremely good designing to do that without an excessive increase in cost.

As to standardization, I have gone the full distance of what I think are the possibilities. There is considerable conjecture as to the probabilities. I believe that there is a fair analogy between the Master Car Builders' standards and those of the motor truck; not that many motor trucks will cross the Continent in a week or two and demand service in San Francisco, but that trucks of a dozen different makes will be built and sent to San Francisco, to Canada, to the South, to Australia, South Africa, India and elsewhere. Why should each truck builder be called upon to maintain his trucks at every point where he sells a truck? Each manufacturer must do this if he is to keep his trucks in service there and hold his business. Why should each one be called upon to maintain wheels that fit no truck but his own, to maintain bearings in those wheels that vary only slightly from those made by the other manufacturers, different steering-pivots, bushings and a hundred and one little details that do not make a particle of difference if they can be picked out from a series of standard sizes when he designs and builds his truck? I am a designer myself and I dislike to be hampered, but I have been up against the practical side of designing in the last several years, particularly from the motor-truck user's side. The user is certainly in a serious predicament when out of touch with satisfactory service parts; it is maddening to find that one cannot replace some particular part because it differs by a few thousandths of an inch from a similar part that is available.

As to claims concerning wheels, I particularly stated that what I have set forth does not prove the case finally. I have used data from carefully made tests, those of a man that intended to equip all his trucks with steel wheels. I realize that all sorts of questions and doubts can arise in the minds of the people who are interested in this matter, because I have done much research work for people who make wood wheels; and it was stated to them several years ago that the steel-wheel claims would not hold water. At about that time I was acting in a consulting-engineering capacity for three different truck builders and this question of wood or steel wheels came up repeatedly. The statements of most steel-wheel salesmen simply cannot stand up against careful analysis. There are some things about wheel design and use that are not fully appreciated in the industry. The same condition exists with regard to a great many other details in a motor truck. But we should know these things. The buyer must know them; otherwise he cannot express his desires. He is the one who pays the bills, not the motor-truck builder.

RUSSELL HOOPES:—Other than the tests described by Mr. Myers, I do not know of authentic data of this kind. As Mr. Myers says, we have heard many statements but have not seen them supported by facts.

We hope that the good work started with the standardization of axles and hubs will be carried on and that the wheel standardization will follow, so that a truck user can stop at any service station and get a new wheel that will fit his axle and hub equipment, and tires that have been standard for many years.

Considering some of the recent remarkable results secured with aluminum alloys, I believe that we shall use aluminum for parts of trucks that heretofore have been under too severe a strain and too expensive. Aluminum will save decidedly in weight, but it may be too expensive to use extensively. In regard to the question of unsprung weight, I hope that in some way additional information will be available.

The paper gives the result of tire tests comparing the cast-steel wheel and the wood-spoke and wood-felloe wheel. The company in which I am interested makes a wood-spoke and wood-felloe wheel of the best possible construction and has found that the wood felloe is a source of weakness. To overcome this it has used a comparatively simple method of inserting a flat-plate washer between the spoke-end and the counterbore in the wood felloe, and has never had the least trouble with this. The same results have been obtained by the Pierce-Arrow company, in its form of spoke sockets or shoes, and also by the White Motor Co. in making its wood-spoke and wood-felloe wheels. The latter company accomplished the same results by using about twice as much timber in the spoke at the felloe shoulder as was necessary to make the head. This gave the spoke ample bearing so that it did not become felloe-bound after hard service. Our company has felt that the metal-felloe truck-wheel has altogether eliminated the wood-felloe.

The matter of chassis lubrication is an essential feature that is commonly neglected. As Mr. Myers shows, an enormous saving can be made by proper practice. Oil is much superior to grease.

Concerning the weights of wheels in connection with unsprung weight, the wood-felloe 36 x 12-in. wheel such as we used to make weighed 216 lb. The weight of the 36 x 12-in. metal-felloe wheel we make is 166 lb. The weight of a 36 x 8-in. metal-felloe wheel, with 2½-in. spokes, is 101 lb.; whereas our type of wood-felloe 36 x 8-in. wheel would weigh 140 lb. In our 36 x 12-in. wheel the metal-felloe makes a saving over the wood-felloe wheel of 50 lb. The metal-felloe 36 x 8-in. wheel saves 39 lb. over the same size of wood-felloe wheel.

These facts demonstrate that the S.A.E. Standard wood-felloe depths have been unnecessarily great since the advent of the metal-base tire that has superseded the fabric base. The tire base strengthens any wheel on which it is placed so much more than the old fabric base that the S.A.E. Standard dual-felloe depths probably could be reduced 25 per cent and the single-felloe depths 50 per cent; in fact, where the single S.A.E. Standard bands are ⅜ in. thick, they probably are strong enough to stand the work without any wood felloe, when a pressed-on tire is mounted.

The wheel weights I have stated are correct, but different wheels will vary in weight on account of the difference in the specific gravity of the various kinds of timber used in the wheel; also, there is a variation in weight of the different pieces of the same rolling of steel used in the felloe band, in the drop-forged sockets and in the plate washers.

Mr. Myers:—In regard to the point that the S.A.E. Standard wood-felloe thickness could be reduced by 25 per cent in some cases and 50 per cent in others, it seems to me that it is advisable to use the thickness specified in the S.A.E. Standards at present, or else use no wood at all. In assembling a wheel, if we have too thin a wood felloe it will not withstand the pressure of the press in which the wheel is put together. It seems to me that it is somewhat risky to advocate cutting 50 per cent off of the thickness of the felloes, and that we should use the present standard until we shall have experimented further with light felloes.

J. H. Wagenhorst:—My experience has been in connection with the substitute for the wood-felloe wheel that Mr. Bachman mentioned. I started on this work in 1913. It consisted of placing steel-felloe wheels instead of wood-felloe wheels on passenger cars, principally on the large-sized No. 66 Pierce-Arrow cars. The result of that development today is that practically all of the wheels that are used on passenger cars now have steel felloes. Our production alone has been about 2,000,000 sets in the last three years, and that may represent one-half or probably somewhat more than one-half of the steel-felloe wheels being used by automobile builders. The Packard, Cadillac, Hudson, Buick, Nash, Overland and Ford companies are using steel-felloe wheels exclusively. Our own factory is making steel-felloe wheels entirely. The wheel itself is considerably lighter than the wood-felloe type, speaking of demountables, ranging from 6 to 16 lb., depending on the size. The average increase in strength is approximately 25 per cent. There is very little change in wood lengthwise with the grain; there is a considerable change across the section. We have taken spokes and kiln-dried them to 4 per cent of moisture, measured them very accurately, immersed them in water for 2 weeks and found, on the second measurement, that there is practically no change in length overall; whereas the cross-sectional change runs as much as ⅛, or possibly ⅞ to 1 in. The crushing strength across the grain is about one-sixth to one-eighth as compared to lengthwise strength.

One of the important questions that was brought up originally in regard to steel-felloe wheels was that of shear-off of the spoke. All of these spokes are supplied with a fillet, which is the natural result in form, and a socket in a steel-felloe base, so we have it somewhat smaller in section, although at the point of shear in the steel it has a considerably greater section, giving a greater endwise area as well as shearing strength.

A MEMBER:—In looking over Fig. 11 of Mr. Myers' paper, I note that truck No. 175 averaged about 0.008 in.; truck No. 172 about 0.017 in.; truck No. 168 about 0.019 in.; and truck No. 162 about 0.026 in. of wear per 1,000,000 commercial ton-miles. Mr. Myers pointed out that truck No. 175 was driven by an extremely careful man. I wonder if there is any significance in connection with that. It seems to me that possibly an analogy can be drawn there between a very careful man's driving and the well-known behavior of a certain car on tires. I know that it is impossible to slip or spin the wheels in starting a certain car from a standing start under any condition that I have encountered. Does he not think that the tire wear is more the result of braking and accelerating than of the load imposed on the tires themselves?

MR. MYERS:—It is not at all fair to compare the wear of the tires between trucks; the conditions were not the same. I give several different variables in my paper. The tests were made as carefully as possible to get some comparative data on wheel service. If the steel wheels were better on account of being more nearly round, it ought to show up as an average on 20 front tires, particularly when we were so careful about the tires. I cannot say that those tires were all exactly the same; I do not believe they were, but I do believe they were very nearly the same. As I pointed out, a comparison of the diagrams shows that all of the tires wore in separate and distinct groups, evidently controlled by the trucks on which they were operated. Truck No. 175 gave the best results; truck No. 172 was next, and truck No. 162 was last. There were marked differences between the worm-drive and the chain-drive trucks on which the rate of wear was greatest, and it is not at all fair to compare the tire wear. What I have given in the paper was just an indication that minimum unsprung weight is by no means all of the story.

Mr. Bachman will acknowledge that as soon as he tries to change the design of anything in his two-cylinder truck very much, he has to change the whole truck. I have seen a change in the size of a bolt in a truck modify 28 other different parts, some of them 6 to 8 ft. distant from the bolt that was changed. The inter-relation of parts in a motor truck is very intimate, the compromises are manifold and good judgment and long experience are needed before one is able to make a compromise. Many conclusions are being drawn from insufficient evidence.

W. B. BUTTERICK:—Has Mr. Myers seen the steel wheel used on the London General Omnibus Co.'s vehicles? It is not by any means so bulky a steel wheel as that which he illustrated. The London General Omnibus Co. operates about 4500 buses on the streets of London and all have steel wheels. I think one will not find a wood wheel manufactured in Great Britain; there is practically none on any truck in London. I find that the London General Omnibus Co. has reduced its tire costs 2d. (4 cents) per mile over those obtaining with wood wheels, since they began using steel wheels. In the case of the omnibus, the wheel is built especially light in weight for that purpose only. The vehicle has an ash frame between two steel clinchers or flitches. Possibly that is the reason why they get a better tire mileage,the resiliency being in the frame not at the wheels.

MR. MYERS:—The wheels used by the London General Omnibus Co. are very light steel wheels, but the difference in the tire cost cannot be attributed entirely to steel wheels. When it changed from wood to steel wheels it also changed a number of other things. The Fifth Avenue Coach Co., in New York City, uses a design very similar to that used by the London General Omnibus Co. It uses a steel wheel that is very well designed and very light.

There is this difference to be considered in regard to the steel wheels used by these two companies and the steel wheels used by the average motor-truck builder. The Fifth Avenue Coach Co. runs its vehicles on a smooth street and straight ahead all the time; but the London General Omnibus Co.'s vehicles run on rougher roads. Some of the streets have cobblestone pavements and the vehicles have more side-thrust to withstand, but not anywhere nearly the amount of side-thrust that a truck in regular service has. The tire-mileage cost of the Fifth Avenue Coach Co. was reduced greatly between the time it used wood wheels some 6 years ago and the present;

but tires have been improved within that time to such an extent that they will give from 70 to 80 per cent greater mileage regardless of the type of wheel; and they have also been reduced in price, so that one cannot get any direct comparison on the wheels themselves. The prime reason so many steel wheels are used in truck construction abroad is that they have great difficulty in getting good wood. It was very difficult to get good hardwood in this Country a few years ago; the Government has requisitioned practically all of the hardwood. However, the Saurer Co., a leading foreign motor-truck builder, still uses wood wheels; some of the wheels illustrated in my paper were made by that builder. That company also uses fewer spokes in its wood wheels than we use in this Country, to obtain the maximum resiliency. The company makes its own wheels. When a wood wheel is made properly, it is superior to the steel wheel, because of its greater resiliency. The French and the Swiss had a sad experience with the steel tire a few years ago, when they used motor trucks over cobblestones. The steel tire produced so severe an impact all the time that buildings deteriorated from great cracks in the foundations, all along the line of travel, and they had to stop using the steel tire. The Saurer Co. appreciated the fact that great resiliency is essential in tires and was the only builder that did not use a steel tire.

Mr. Butterick:—Why do not the White and the Pierce-Arrow companies use the wood wheel? I believe they said they could not get any suitable wood.

Mr. Myers:—That is exactly the Pierce-Arrow company's reason. This company made its own wheels and they were fine ones. Records showed wheels that had been in service from 1 to 5 years in the Arizona and New Mexico districts where some wheels dry out in a few months and go to pieces, but the Pierce-Arrow company's wood wheels had practically a perfect record of service. But those wheels were properly dried, finely finished, excellently fabricated and thoroughly painted. Two coats of primer were put on them as soon as they were fabricated and before the wheel went out for test. Before that truck got away from the plant it had five coats of paint. It was a good truck and the man who bought it took pride in it and painted it from time to time. All material must be used with respect to its characteristics, and moisture must be kept out of a wood wheel. It can be done without any trouble if one recognizes that need and cares for it. I seem to be a champion of the wood wheel, but I do not want to take that attitude. I am trying to face facts. Let the man who has the facts of the matter in his possession present them.

J. E. WOLFF:—Very little good hickory is grown abroad. I believe this is the chief reason that more wood wheels have not been used in London. Aside from that, we have supplied several hundred sets of wood wheels to foreign companies. These have been very satisfactory; there has not been one complaint; so, there are wood wheels abroad as well as steel wheels.

ERWIN L. SCHWATT:—Mr. Myers' paper states that "the average rate of wear of the rear tires on the chain-drive trucks was 53 per cent in excess of those on the worm-drive trucks." It seems that, when the chain is under tension, the wheels will slip a little when the loaded frame changes its position relative to the axle due to the roughness of the road.

MR. MYERS:—The figures are not truly comparative between trucks, or for unsprung weights; but if a man says that the truck with the light axle will show decreased tire wear, how does he reconcile that statement with this condition where it does show increased tire wear? They did not measure any clutch performance or check up on the drivers. They sent each driver out on a truck equipped with a wood wheel on one end of each axle and a steel wheel on the other. The driver did not know what was going on at all, and operated the truck just as he thought he ought to operate it.

MR. SCHWATT:—That skidding action probably is the reason for the excessive wear, and the relative weights of the rear ends do not enter into it. In the case of the worm drive we have the possibility for a skid, and also in the chain drive there is an actual skid.

MR. MYERS:—That *might* be true, but that is just one of a number of variables that can enter into this matter. How can anybody make flat statements that this wheel or that wheel can accomplish all these economies when so many other variables can more than offset any saving?

MR. BUTTERICK:—The Albion truck people were using the method of automatic lubrication with a wick in 1908.

MR. MYERS:—Possibly. I believe, however, they did not make substantial magazine brackets, but used oil-cups which are in the nature of excrescences. When one is knocked off it is often forgotten that any oil needs to be applied at that point.

SOME REQUIREMENTS FOR THE RAIL MOTOR-CAR[1]

By W L Bean[2]

The rail motor-cars now used by the New York, New Haven & Hartford Railroad are illustrated and commented upon, and statistical data regarding their operation are presented. The features mentioned include engine type and size, transmission system, gear-ratio, double end-control, engine cooling, heating by utilizing exhaust gases and exclusion of exhaust-gas fumes from the car interior. A table gives revenue data.

The usual steam-railroad coach is heavy. The impacts of that car on the track are sufficient to cause a yielding of the roadbed, track and ties; whereas a light vehicle has little or no effect. The smaller rail-pressures reflect themselves, for instance, in the fact that we have trouble in operating electric crossing-signals, particularly on Monday morning when there has been no traffic over the line on Sunday. That is not altogether due to weight, it is a matter of wheelbase and speed; but it shows that the wheel pressure on the rail is so light in relation to what usually operates over the tracks that it is relatively negligible. That has its effects on the riding qualities, yet we must keep the vehicle light to propel it economically with the gasoline engine.

We think the rail motor-cars now used on the New York, New Haven & Hartford Railroad constitute an exceedingly good beginning and that, as a foundation from which to work, a much better unit can and should be developed. But there is no use in disguising facts. If one will ride in one of those cars for a number of hours, one must admit that the wear and tear on one's nerves is more than in riding on a steam car; and the patrons bring up that proposition. So, while we are working for more power, we need the greater flexibility together with perhaps a six-cylinder engine, because I believe that we should not consider more than six cylinders for some time to come. We must get a six-cylinder engine that is designed for rail motor-car service; this means continuous heavy duty and not a light throttle and drifting, slowing up for traffic or turning corners. It means full throttle the greater part of the time. The engine must be designed for that usage not only from the angle of ability to stand up but with regard to minimum vibration, which of course includes the suspension of the engine. That is important in a rail motor-car because of

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[1] Metropolitan-New England Sections paper.

[2] Mechanical assistant to the president, New York, New Haven & Hartford Railroad Co., New Haven, Conn.

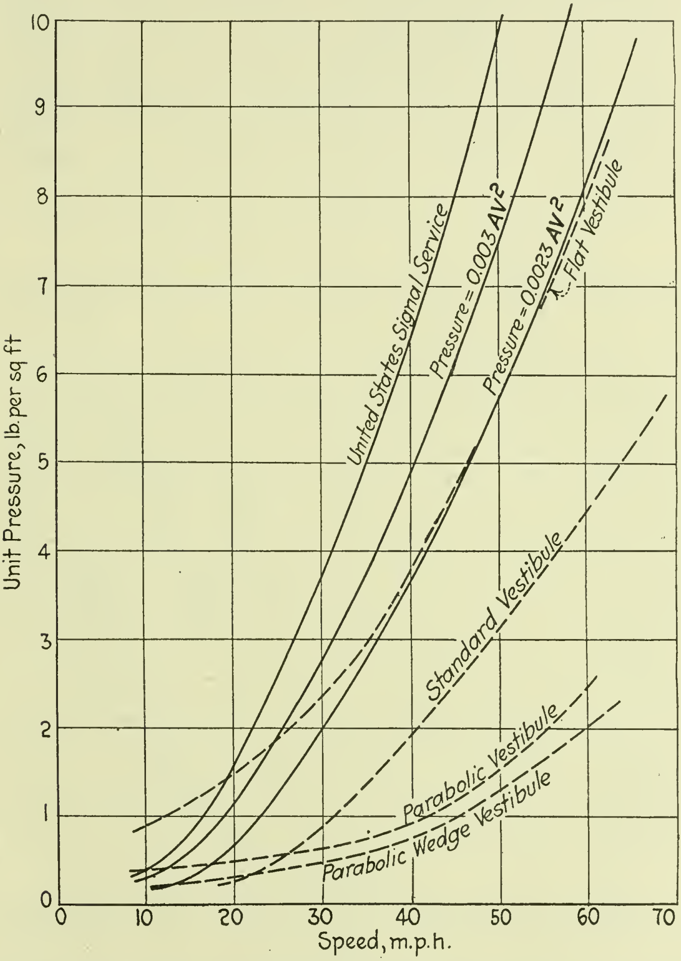

FIG. 1—WIND-RESISTANCE CURVES FOR CARS OF DIFFERENT FRONTAL SURFACES OPERATING AT VARIOUS SPEEDS

the sort of drumhead effect given by the roof, floor and sides of the car.

Transmission systems should be designed to permit operating the engine at less than normal speed; that is. at favorable speeds for economy and quietness, when the demand on the engine is less than normal. For instance, a car may drift successfully for 15 to 20 miles down a water-grade. That should be done through a gear-ratio properly adapted; we should be able to let that car drift at less than the full engine-speed. The car may require only power enough to propel it from 20 to 30 or 35 m.p.h. but, if its engine must turn over just as fast as if it were making 35 m.p.h. on the flat, developing a corresponding horsepower, we get conditions of considerable vibration. I think that can be avoided in a measure

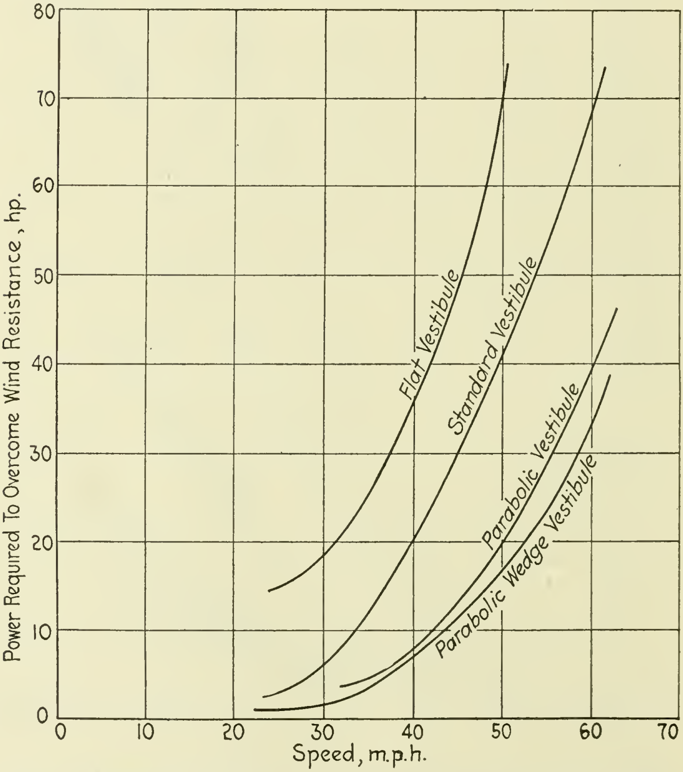

Fig. 2—Horsepower Required To Propel Cars Having Different Contours of Frontal Surface at Various Speeds

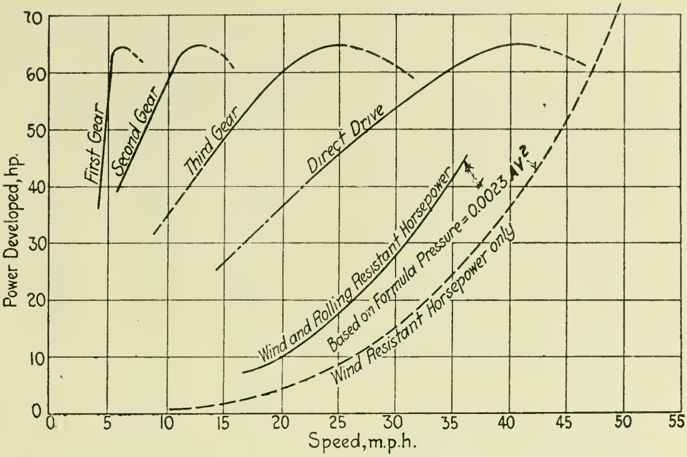

Fig. 3—Relation between the Horsepower Developed by the Four-Cylinder Mack Engine and the Car Speed

through the use of proper gear-ratios, and we intend to conduct further experiments with our cars in that direction.

As to the double-end car, I wish we could consider developing cars now without that feature; but the operating requirements in some localities are such that a double-end car would be much superior, from an operating standpoint, to a single-end car. We have terminals on our railroad where a car that could come in and shuttle out would be vastly superior to one that would have to go out to a turntable or a wye; in fact, it would be almost impossible to maintain some of the schedules that are contemplated for such a double-end car, if a single-end car were used.

The matter of cooling the engine requires special consideration. We find that heavy, continuous service requires a greater ability to dissipate heat than does that of highway cars. The matter of heating by utilizing exhaust gases efficiently and satisfactorily demands considerable study and arrangements that will keep gas fumes outside of the car body are necessary and have not altogether been worked out.

Table 1 gives data covering the revenue service up to a recent date. Incidentally, the car had made considerable mileage before that. Under the heading Average Passengers Per Trip, the total average of non-revenue and revenue is 28.5 for Car No. 9000, 23.4 for No. 9001 and

TABLE 1—STATISTICAL DATA ON GASOLINE-DRIVEN RAIL MOTOR-CARS

	Car No.		
Items	9,000[a]	9,001[b]	9,002[c]
Placed in Service, 1922	Jan. 4	Jan. 18	Jan. 30
Daily Total, miles	146	59	139
Total Mileage to May 6, 1922	12,441	5,479	10,396
Number of Revenue Passengers to May 6, 1922	11,115	6,822	11,013
Average Number of Passengers per Trip			
Revenue	20.0	19.6	39.6
Non-Revenue	8.5	3.8	3.6
Total	28.5	23.4	43.2
Total Number of Trips	533	366	301
Total Delay during Period, min.	213	340	179
Total Delay per Trip, min.	0.4	0.9	0.6
Number of Stops per Day	42	12	25
Car Trips Replaced by Steam Train	3	10	25
Number of Trips per Day	6	4	4
Average Speed, m.p.h.	23	25	20

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[a] Operated between Derby, New Haven and New Hartford, Conn.

[b] Operated between Fairhaven and Tremont, Mass.

[c] Operated between Litchfield, Danbury and Waterbury, Conn.

43.2 for No. 9002. Those are averages per trip, not the maximum on the car at any one time. Therefore, it will be noticed that two of the cars are handling on the average from 3 to 12 people less than the nominal seating capacity of the car, yet at times the cars are badly crowded. The delays per trip are 0.4, 0.9 and 0.6 min. The average speed in miles per hour is shown at the extreme right; in most cases the cars run on an average from 35 to 38 m.p.h., because of the number of stops they have to make. I rode in one car at 42 m.p.h.; but that is too fast for comfortable riding, so far as engine vibration is concerned.

Figs. 1 and 2 show studies that were made of wind resistance and are not intended to be more than an attempt to show in a rough way the relation between wind resistance and the different contours of the frontal surface. Fig. 3 shows the horsepowers developed and their relation to the speed with the four-cylinder AC Mack type of engine. The vertical distances measured between the horsepower curves represent the rolling-resistance approximately, and it is important to note the rapid rise in proportionate power resulting from the engine, because of the fact that, in the unit car, the relation between frontal area and weight is very different from what it is in a steam train. The problem of wind

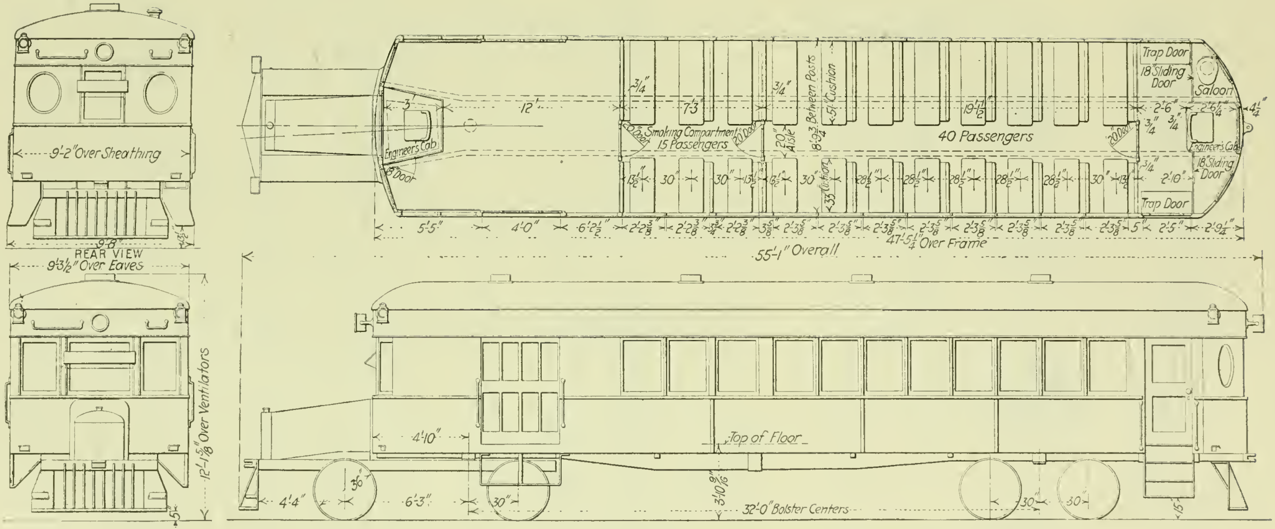
9'-2" Over Sheathing
9'-8"
REAR VIEW
9'-3½" Over Eaves
12'-1⅝" Over Ventilators
Engineer's Cab
Door
12'
Smoking Compartment 15 Passengers
20" Aisle
8'-9¾" Between Posts
51" Cushion
33" Cushion
40 Passengers
Trap Door
18" Sliding Door
Saloon
47'-5¼" Over Frame
55'-1" Overall
Top of Floor
4'-10"
4'-4"
6'-3"
30"
36"
3'-10 9/16"
32'-0" Bolster Centers
15"

Fig. 4—Plan View of a 55-Passenger Motor Railcoach

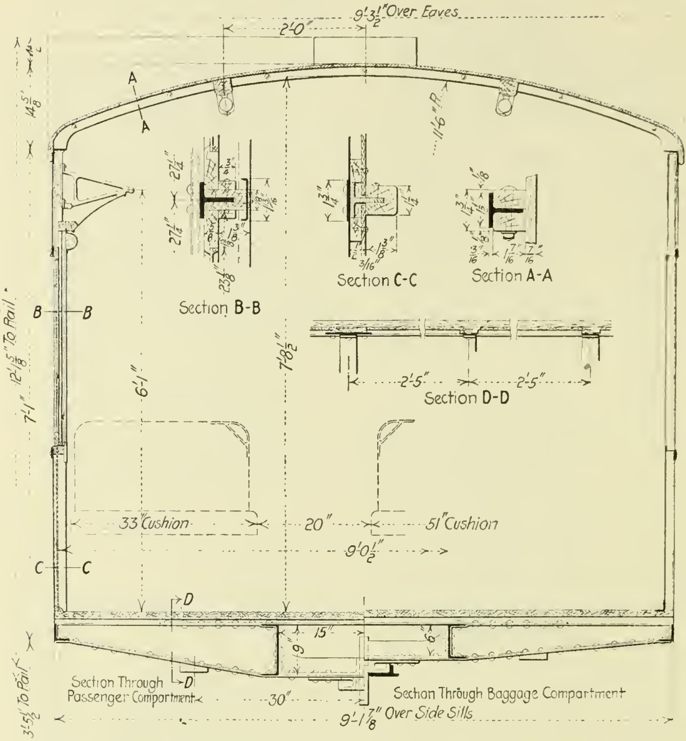
9'-3½" Over Eaves
2'-0"
14⅝"
11'-6" R.
A
A
Section B-B
27¼"
27¼"
23⅛"
Section C-C
1¾"
1¼"
3/16"
Section A-A
1¾"
7/16"
Section D-D
2'-5"
2'-5"
B
B
12'-1⅝" To Rail
7'-1"
6'-1"
7'-8½"
33" Cushion
20"
51" Cushion
9'-0½"
C
C
D
D
15"
6"
3'-5½" To Rail
Section Through Passenger Compartment
30"
Section Through Baggage Compartment
9'-1⅞" Over Side Sills

FIG. 5—CROSS-SECTION OF THE BODY OF A 55-PASSENGER MOTOR RAILCOACH

resistance is real in unit-car resistance, whereas it is decidedly minor in steam or heavy electric types.

Fig. 4 shows approximately our idea of the floor area and arrangement of the larger car that we feel would cover a very substantial field. That car would seat 15 passengers in the forward compartment, which would be in the rear of the baggage compartment, and the seats would be used by the smokers; in the rear of that there would be seats for 40 other passengers.

Fig. 5 is a cross-section of the car body, showing the lightness of construction that one must adopt in designing bodies in order to keep within the power limitations of gasoline engines. At the same time it would give some insulation that would help solve the heating problem and eliminate vibration and noise.

[The discussion of this paper is printed on p. 609.]

AUTOMOTIVE RAIL-CARS AND FUTURE DEVELOPMENT[1]

By L G Plant[2]

The many improvements effected in gasoline-engine construction during the war for airplane, heavy truck, tractor and tank usage have done much toward making the gasoline-driven rail motor-car a practical possibility today.

The gasoline-electric cars built by the General Electric Co. are mentioned and light rail motor-car construction is discussed in general terms. Reliability and low maintenance cost are commented upon briefly, and the requirements of service for rail motor-cars are outlined.

From the start, developments in automotive engineering have inspired attempts to adapt the same principles of propulsion to railroad cars. There always has existed a field for equipment of this description, due to the fact that the operation of a steam locomotive and a train of cars involves certain elements of cost that cannot be curtailed in proportion to the size of the train; so, in passenger service, light local steam-trains have been operated at an expense that is excessive in proportion to the revenue received. It is only recently that the most vital factors contributing to the commercial success of the automobile truck have been applied to the construction of self-propelled rail-cars, but within the past year developments in this field have been moving rapidly toward a type of self-propelled car that can be substituted successfully for a steam-train in certain classes of service. No recent development in railroad equipment has aroused such universal interest on the part of manufacturers and nearly all railroad departments within so short a space of time. The successful adaptation of automotive principles to railroad cars will, I believe, prove a very great benefit to the railroads in enabling them to reduce the cost of light local passenger service and increase their gross revenue by augmenting and improving the character of this service.

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[1] Metropolitan-New England Sections paper.

[2] Associate editor, *The Railway Review*, Chicago.

In view of these circumstances, the question arises as to why previous developments in this direction have not met with more permanent success, and it is still something of a mystery why so obvious a solution of the problem as is found in the modern rail motor-car should not have been discovered earlier. But, before discussing the more fundamental causes that retarded this development, it is pertinent to say that our railroads have never been more severely pressed to devise operating economies than within the last year and have never been more keenly alive to the possibilities of any equipment designed to reduce operating costs. Coincident with this attitude, the situation with the truck builders has also been propitious for progress in this direction.

During the period of the war, self-propelled rail-car construction came to a standstill, and many of the cars previously purchased by the railroads were withdrawn from service, due to their high maintenance-cost and unreliability in operation; but, while the war apparently retarded development along this line, in reality the many improvements effected in gasoline-engine construction, designed not only for airplane use but for heavy trucks, tractors and tanks, have done much toward making the gasoline rail motor-car a practical possibility today. In distinction from the relatively slow-speed heavy engines used originally in the McKeen and Hall-Scott cars, we now have in the heavy-duty truck-type of engine a very much lighter high-speed engine capable of exerting a high torque through a wide range of speeds. This has an important bearing on rail-car construction, not only on account of the reduced overall weight of the engine but because the reduced weight of the reciprocating parts obviates the difficulties occasioned in the earlier cars by the inertia of the heavy engine parts. This will account for some of the difficulties encountered on the earlier types of rail car which, although they may appear somewhat crude in the light of present-day practice, represented a real mechanical achievement at the time of their construction.

This is illustrated best in the construction of the gasoline-electric cars built by the General Electric Co. involving the design of a special gasoline engine with two sets of four cylinders each, forming the V-type arrangement that has since been used extensively and indicating that, at the time these cars were first built, they embodied the most advanced engine construction available. Now, however, the variable-speed characteristics developed in engines of the type now considered for gasoline-rail-car operation, together with the variable speed-ratios in the transmission mechanism employed with these engines, afford an element of flexibility in speed control that obviates the necessity for the interposition of electric drive from the standpoint of speed control; and it is believed that the additional weight and first cost involved in an electric generator and motors preclude the economical use of this form of transmission in rail motor-cars of the type now under consideration.

LIGHT RAIL MOTOR-CAR CONSTRUCTION

It is apparent, therefore, that from the standpoint of the motive-power unit, the type of rail motor-car now discussed is fundamentally different from that developed prior to the war and represents a distinct advance over these earlier cars. But while the development of light high-speed gasoline-engines capable of operating continuously under heavy loads has been advancing rapidly, there also has been under way a development, inspired partly by what the automobile builders have accomplished through the use of alloy-steels and also by the trend in street-railway-car construction toward lighter weight, such that the builders of this equipment are now able to design very light cars for operation where the power limitation is severe. One of the most remarkable examples of this construction is a double-truck car-body weighing 11,000 lb. that has seating capacity for 46 passengers. This car is 42 ft. long and has a baggage compartment. Broadly speaking, therefore, it can be said that the modern rail motor-car is the embodiment of an improved powerplant and refinement in car construction.

While it is believed that in the numerous designs of self-propelled rail-car in service or under construction there is available to any railroad a type that it would be justified in buying at present, it is admitted that the design of these cars is still in a progressive state principally with respect to details of construction that will insure their reliability and low cost from a maintenance standpoint, and that will also increase the capacity of the equipment. It is not possible to determine from the figures now available the largest number of passengers that can be handled more economically in self-propelled rail-cars than in a steam-train and, of course, this figure would depend upon local conditions, but it is safe to say that under ordinary circumstances, the operation of self-propelled cars with as many as 80 passengers would show a considerable saving over that of a steam-train carrying the same number of passengers.

Although the question of using trailers or a single large car would need to be decided in this connection, this is not regarded as fundamental to the solution of a problem that, in reality, lies in the design of a motive-power unit of sufficient capacity without sacrificing any of the characteristic features in commercially available types. From a theoretical standpoint, the use of a motor car and trailers in place of a single large car will increase the frictional resistance and dead-weight per passenger slightly; but, practically, operating conditions peculiar to the railroad on which this equipment is operated will prove the determining factor, so that it would be a mistake for any manufacturer who is looking toward the development of greater carrying-capacity in this type of equipment either to depend entirely upon the use of trailers or to commit himself to a design that would preclude the use of trailers.

The application of more power to self-propelled cars presents a real problem, since there are few commercially available engines of the type adapted to this service that exceed 60 hp. at normal speeds. It is in this connection that the unit steam-car has a unique advantage, since it is capable of developing as much as 300 hp. with a flash type of boiler. While it is understood that the unit steam-car has some very special advantages in connection with the subject of self-propelled cars, it is recognized that this type has reached a more advanced stage in relation to its ultimate development than the gasoline-engine car, so that further discussion of this subject will be confined to the latter type which must still be regarded as being in a formative stage.

With gasoline engines of 60 hp., the best that can be anticipated appears to be a car that will seat approximately 40 passengers, carry baggage and operate normally at a speed of about 40 m.p.h. To effect any considerable increase in the size of this car or render it capable of pulling a trailer at the speeds required in main-line service will necessitate more power, either through the use of a larger engine, which ordinarily involves special and expensive construction, or the use of two engines, which involves certain special problems in their control. Assuming that it were practicable to design an individual transmission of sufficient flexibility to enable the simultaneous operation of both engines, and that it were possible to control the operation of these engines satisfactorily, the use of two 60-hp. engines would have a theoretical advantage over a single engine of larger capacity since, whenever the power requirements drop to the capacity of one engine, it would be possible to run a single engine at full capacity and thus realize more efficient operation than when a larger engine is operated at a fraction of its capacity. The gasoline consumption of a rail motor-car seating 40 persons and carrying baggage will approximate 0.2 gal. per mile, and it will be desirable to maintain a proportionally economical rate in larger cars. Another factor that should encourage development in the direction of using two engines is the element of reliability afforded by two independent driving engines since, ordinarily, one engine will continue to operate should the other fail. Probably no other factor proved more discouraging to the successful use of both the McKeen and the General Electric Co. cars than the numerous engine failures that were encountered in the operation of this equipment.

RELIABILITY AND LOW MAINTENANCE COST

Reliability and low cost from a maintenance standpoint as already referred to undoubtedly constitute the most difficult problems with which designers of this equipment have to contend, since they are matters in which the railroads are most exacting. Maintenance of cars and locomotives already costs the railroads as much as either train wages or locomotive fuel, and any failures in train service add to this expense and the difficulty of operation. The problem is complicated further by the fact that equipment of the type under consideration often could be operated to the best advantage between points that are isolated from shop facilities. While the substitution of a hard and smooth rail would seem to facilitate the adaptation of automotive equipment to rail service, the absence of that element of flexibility afforded by a resilient tire operating over the ordinary highway surface in reality makes the adaptation of automotive equipment to rail service a more difficult problem. Not only do uneven joints and cross-overs introduce more severe vertical shocks, but abrupt changes in the alignment of the rail, as at switches and on curves, cause far more severe lateral shocks than are ever encountered in highway service.

Moreover, the absence of any element of elasticity between the engine and the driving-wheel tread, as provided in automobile construction by a resilient tire, operates against the efficiency of the gasoline engine that can be operated only to the best advantage when the transmission is capable of absorbing the ordinary pulsations of the engine and cushioning the shocks occasioned by any abrupt variation in the speed between the engine and the wheel tread.

For these reasons it is believed that the most successful development in this class of equipment will tend toward the standards in truck, axle and wheel construction that many years of railroad service have demonstrated to be safe and economical; that the use of rotating axles mounted in special journal boxes fitted with frictionless bearings will become general; and that the most desirable form of transmission will prove to be one that is flexible with respect to the vertical and lateral blows transmitted through the driving-wheels. But in whatever development work is undertaken, either in this direction or with respect to enlarged engine capacity, it is safe to say that the most successful results will be obtained where the experimental work is conducted in conjunction with the railroads.

Finally, in discussing the future of the self-propelled rail-car, the question of greatest importance to the manufacturer and of some moment to the railroads as it may affect the cost of this equipment, is the matter of production which, in turn, depends upon the prospective field for this equipment. It is not unreasonable to assume that in short-line railroad-service, wherever it is possible to dissociate freight-car movement from passenger service, the gasoline-driven rail motor-car will supersede the steam passenger-train eventually. The rate at which this transformation will take place will depend not only upon the finances of the railroad, but upon the attitude of the manufacturer toward financing this purchase. Reliable figures are available to show that wherever cars of this description have been operated by the independent short-lines, they have reduced the cost of operation in comparison with steam service; and wherever the road was incurring a loss with steam service, the gasoline rail motor-car enabled it to make a net profit. Moreover, there are now available several types of car admirably adapted to any class of passenger service ordinarily operated by the short-line railroads.

Trunk-Line Requirements

Turning to the trunk-line railroads, the question cannot be answered so easily; first, because the railroads do not themselves know the extent to which self-propelled rail-cars can be substituted profitably for steam service and, second, because types designed to carry more than 50 passengers at the desired speeds are not yet available.

It is safe to say that there is an immediate field for possibly 1500 cars of the types already available. It is understood, of course, that many railroads will want to delay the purchase of these cars pending the development of new types, while others will want to observe the operation of this equipment on other railroads before committing themselves. Also, the question of financing the purchase of these cars will come up for consideration and it is reasonable to believe that, once a depreciation rate on this type of equipment has been determined reliably, some form of equipment trust applying to a number of these cars would facilitate their purchase. Altogether, the development is yet so new that it may be several years before we can look for any volume of purchases that mean the production of these cars on a large scale, despite the fact that a great field of usefulness awaits them.

THE DISCUSSION

R. B. Abbott[3]:—I am chairman of a committee of the Philadelphia & Reading Railway Co. on the question of the rail motor-car for use on branch lines. After we determine what type is best suited to our purpose, we will then make a study of all cars that approximate this particular type.

A car to suit our purpose must be capable of being used as a motor car with a trailer or, perhaps, with another car in addition to the trailer, because the demands on our branches vary so much on different days in the week and at different times of the day that a single unit car would not of itself solve many of our problems. The car must also be reasonably comfortable and not noisy or oil-smelling. It should, we think, be capable of control from either end, so that it will not be necessary to turn the equipment at the end of each run.

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[3] Assistant general superintendent, Philadelphia & Reading Railway Co., Reading, Pa.

G. C. HECKER[4]:—When the electric railway people first adopted light-weight cars, they worked on the theory that they would not replace the seats, seat for seat; that is, they would not attempt to give exactly the same service that had been given previously with large double-truck cars but, rather, increase the service considerably by running cars on closer headway. The people in the different communities found after they had overcome their first dislike for these cars, which, perhaps, were not so comfortable as the large double-truck cars, that they were really getting much better service. On a branch line of a steam railroad where very infrequent service is now being given by steam operation, it might be possible, with the gasoline rail motor-car operating in single-unit light-weight cars, to give a very much improved service. I believe the public can be made to realize that they will get very much better service if they will put up with a little less comfortable car. As the gasoline rail motor-car is developed, there unquestionably will be many refinements that will reduce the objections of the riding public to this form of transportation.

J. E. BURRELL[5]:—The Pennsylvania Railroad has a committee that has been investigating the various types of rail motor-car that are in service at a number of points. The company does not operate any cars of this type on its lines. One car, however, is operated on a branch line by another company. The car is similar to those used by the New York, New Haven & Hartford Railroad, and has been giving very good satisfaction. We are, of course, somewhat in the same position as the Philadelphia & Reading; we are trying to find the car that will suit our purpose best and, after ascertaining what car that is, we probably will install it on the line.

ARTHUR J. SCAIFE:—It is our understanding that the great need today is for some kind of combination baggage and passenger car that will take the place of the present equipment used on many short-line railroads where it is necessary to use a passenger car, a baggage car and a locomotive, with a full train-crew. This equipment cannot be operated without a loss and the company usually runs one train a day because it is required to do so.

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[4] Special engineer, American Electric Railway Association, New York City.

[5] Superintendent of passenger transportation, Pennsylvania System, Eastern region, Philadelphia.

Very little work has been done on rail motor-car equipment by our company, and that has been only within the last few years. We are trying to find out first just what the requirements are with reference to seating capacity. It will be necessary to go at this proposition with an open mind. The thing that we have run up against is that men have been thinking in railroad terms. They immediately criticize a rail motor-car job and ask how it compares with the present railroad equipment and Master Car Builders' standards. If the automotive rail-car builders and the railroad operators go at this problem with open minds, I believe that something can be accomplished.

L. G. NILSON:—The present International car, which has a good appearance and is doing very well, naturally has the earmarks of the ordinary motor-truck. I believe that when we consider the larger sizes that the railroads undoubtedly want, we shall come back to something like the McKeen car; that is, the power and transmission, the whole drive and equipment should be on the forward truck or at least on one truck. The car body proper could be made very light, with a light trailer, arranged so that it could be uncoupled in a very few moments. In that way the power unit could be gone over at regular intervals once a week, or even inspected once a day, and it would not be necessary to tie up the car body. The car bodies could be run 24 hr. per day or as long as desired.

I believe we shall find that a driving unit of this kind equipped with spur gears will give better satisfaction than one equipped with bevel gears. The bevel gears are doing very well in ordinary sizes, but the use of too large sizes causes many difficulties, not so much on account of the gears as on account of the mounting. Unless the mounting and the housings are very rigid, the teeth simply tear themselves out; with spur gears, there is less trouble of that kind.

I would like to predict that we shall see an internal-combustion engine almost as elastic as a steam engine in its action, possibly within 3 years. Then the problem of transmission and control will become very much easier.

HENRI G. CHATAIN:—Some 20 years ago, Mr. McKeen induced E. H. Harriman to invest some money in the construction of a rail motor-car that had a mechanical drive. At about the same time I persuaded the General Electric Co. to engage in the construction of gasoline-electric rail motor-cars. Mr. McKeen built some 150 to 200 rail motor-cars and, I believe, they are in successful operation today. The General Electric Co. built 100 rail motor-cars and approximately 98 are in operation to-day.

I have listened with great interest to the gentlemen who have spoken in regard to the number of miles that the newer types of car are making. It is interesting to know, and I think the information is correct, that one of our cars recently completed 1,000,000 miles in service, and there are a number of them that have gone over 600,000 miles.

At the time we began to build rail motor-cars, if we had had superintendents on the railroads who would listen to arguments in favor of light weight and not insist upon having many things hitched to the car that belonged to the steam locomotive, we would have built small light cars a number of years ago. But the railroad representatives could not agree with such a viewpoint, and I am not sure that they can agree today. Each superintendent wants a different kind of car. Some want trailers; some want to go faster, and others want to go slower. All of this involves differences in design and attendant high cost of production.

Mr. Bean studied the proposition of the light-weight rail motor-car and has convinced me of its merits. He is willing to do without couplers and many other things, provided he gets a good and safe rail motor-car.

I am not an advocate of the mechanical drive. I followed the McKeen rail motor-cars very closely and have the facts and figures covering thousands of miles of their operation and also similar data for the cars built by the General Electric Co., covering a comparable number of miles of operation. The mechanically driven car will operate on less gasoline per mile, but it will cost more for maintenance and repair, because it does not possess what I like to call "squashiness." It is not an automobile with rubber tires, but runs on rails that are not flexible from the transmission viewpoint. No engineer has yet developed ways and means of attaining suitable flexibility between the engine and the track, and this is the important factor so far as upkeep is concerned.

The gasoline-electric drive has four points of advantage:

(1) The engine can be loaded at all times. As the gas engine is governed by changing its compression, it is obvious that if it can be loaded properly

at practically all speeds and all through its operating range, the efficiency can be increased by increasing its average working compression

(2) It is more flexible. The speed changes blend from one to the other because of the nature of the electrical units employed

(3) The prime-mover can be operated at a speed below its normal rate and yet maintain a high car-speed. The high engine-speed is maintained during the accelerating period and for grade work, but is reduced during the period of free running on the level or down a slight grade. These conditions can be well taken care of by the gasoline-electric drive

(4) It makes possible double end-control

I shall not take any very decided stand on whether the transmission should be capable of a complete conversion of energy or not. I think that there are a number of transmissions that do not completely convert the energy. They possess all the desirable features such as the flexibility, loading and the various speeds of operation of the engine, with but small losses of energy. The installation of an engine in a rail motor-car is an extremely difficult thing. It cost us a large sum of money before we found out how to do it. A combination of felt and springs seems to be the most effective; we are using it and it has been reasonably successful.

We built eight-cylinder engines, but they are not so desirable as those of six cylinders or multiples thereof.

The position of the exhaust is an important matter. Mr. Bean points out that there must be some means of preventing gaseous odors in the car, because they are very disagreeable to passengers. We tried every conceivable position for an exhaust and found that the best place to put it is directly overhead, with not too much muffler, so that the gases will go up as high as possible due to their velocity. Exhaust at the rear of the car tends to roll up and come in through the back windows.

The greatest need in the rail motor-car field today, to make it an economic as well as a manufacturing proposition and therefore desirable for both the manufacturer and the user, is the standardization of requirements. This probably can best be brought about by the Society working in conjunction with railroad representatives of authority.

Charles O. Guernsey:—Gasoline-propelled motor-coach equipment should be only of such size as can be operated with commercially proved engines and handled by a crew of two men. With larger engines that have not been proved out in severe duty and under commercial conditions, we may get into some mechanical difficulty. For cars larger than can be handled by two men, the saving in cost will not be sufficient as compared to steam equipment to justify the use of gasoline-propelled cars. Like any other broad statement, this is undoubtedly subject to some limitation.

Generally speaking, the gasoline engine of more than 5-in. cylinder-bore has not been proved in commercial automotive service. It is true that large engines have been used in various installations, such as aircraft or private yachts, but for such service the first cost, operating cost and maintenance are not of prime importance. Engines of this type would not be satisfactory in motor-coach service. If we assume then a 5-in. cylinder as being the maximum that can be used safely, we are confronted immediately with a limitation of horsepower dependent upon the number of cylinders that are used. Four-cylinder engines of this size have been well proved and undoubtedly will be successful in this service. It is possible that some six-cylinder designs which are now on the market may also be successful. For larger powers, nothing has been developed as yet, and I doubt whether there will be a sufficient demand to justify the development of 8 or 12-cylinder engines for this service, to say nothing of the complications incident to such a multi-cylinder design. If the foregoing assumptions are correct, we are limited in the case of a four-cylinder engine to about 70 hp. as the maximum that is available; or, in case we accept the six-cylinder design of about the same cylinder bore, we can expect to get about 100 hp.

The designer of these cars should bear in mind that the car must represent a combination of automotive and railroad practice. The railroad standards as to safety, comfort, steadiness of riding and low cost of maintenance and operation must obtain. Because of the limited power available, the weight must be kept to a minimum. This indicates, therefore, the use of alloy-steels, light-weight designs, anti-friction bearings and the like, as customarily used in automotive practice and as already proved in such service. The weight of car that can be handled satisfactorily with the engines of the powers mentioned will depend, of course, upon the speed required, the road conditions, the number of stops and the acceleration that must be had.

In general, it is my opinion that, with the four-cylinder engine, the outside limit of weight for general all-round satisfactory performance is about 18 tons loaded, and for the six-cylinder engine the outside weight should not exceed from 22 to 25 tons. In a properly designed car the four-cylinder engine should handle about 45 passengers in combination with a baggage or express load of about 1 ton satisfactorily. The six-cylinder engine probably would handle a passenger load of from 55 to 60 passengers in combination with about 2 or 3 tons of baggage.

Demonstrations of a four-cylinder railroad motor-coach developing 61 hp. show the results given in Table 1.

TABLE 1—ACCELERATION OF A FOUR-CYLINDER RAILROAD MOTOR-COACH[6]

Acceleration from a Standing Start to Speed, M.P.H.	Time Min.	Time Sec.
25	..	30
29	1	..
35	2	..
41	2	40

[6] Light weight of car, 13 tons; loaded weight, 17 tons.

The gasoline consumption varies from 5.2 to 7 miles per gal., depending upon the conditions. The normal speed of the coach at the rated speed of the engine is 35 m.p.h. and the maximum speed with full load is 48 m.p.h. The operating cost, including a crew of two men at standard wages, gasoline, oil, maintenance, depreciation and interest on the investment, is about 29 cents per mile. This figure will, of course, vary with the local conditions.

W. G. BESLER[7]:—In my opinion, there are certain places where a gasoline-propelled vehicle finds its proper application in railroad service, but in those cases where a cement highway costing from $40,000 to as high as $120,000 and in some cases even more per mile, is constructed at public expense, paralleling a railroad, why should branch-line service, which is the only place where a gasoline rail-car finds its proper use, be continued? In such an instance the railroad company had better stop operating, invest its money in motorbuses and continue service upon a highway provided for it free of expense, than subject itself to the burdens of expense for rails, which require renewal, maintenance, supervision and all that goes with railroad service.

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[7] President, The Central Railroad Co. of New Jersey, New York City.

GEORGE L. SHINN[8]:—Our designation of the practical application of the gasoline-driven rail-car at present is that it can be used with great advantage for light traffic conditions. I say this because on our road we have substituted a White combination passenger-and-baggage car where we formerly used a steam locomotive and two passenger coaches to handle the traffic. We have not yet arrived at a definite figure but a conservative estimate indicates that, by the use of this gasoline-driven car, we shall effect a saving of $15,000 per year. We believe that, by the use of this gasoline-driven rail-car, we are giving service superior to that formerly rendered, and we note from the expressions of opinion that have reached us that the patrons on the line are much better satisfied. We maintain the same schedule as when the steam trains were in operation and find that we can give even greater service should it be necessary and desirable.

The car that we have in operation is governed for a maximum speed of 33 m.p.h. and, due to its excellent acceleration and easy handling, we are maintaining the former steam-train schedule without difficulty and could, if desired, stiffen this schedule. We have no hesitancy in saying that our experience with the gasoline-driven rail-car is very satisfactory in every way.

J. W. CAIN[9]:—In making our investigation of gasoline-propelled rail motor-cars for the member lines of the American Short Line Railroad Association, we did not go into the subject technically but, instead, considered the different cars more from the standpoint of practicability as evidenced by actual service. We approached the subject from three different angles and our final conclusion was based on a summation of the information thus received.

Our membership consists of some 500 different railroads located throughout the United States, and for a great many years they have been the proving grounds for the different rail motor-cars brought forth. Indeed, there has seldom been a gasoline-propelled railroad-car built that has not at some time or other found its way to one of these properties. We, therefore, had a source of extremely valuable information and sent to each of these lines a questionnaire, of which the following are the principal questions:

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[8] President, Pennsylvania & Atlantic Railroad, New Egypt, N. J.

[9] Manager of purchases, American Short Line Railroad Association, City of Washington.

(1) Are you using motor equipment on your line, and if so what make?

(2) How long has it been in service?

(3) What is your average operating cost per train mile?

(4) Approximately what mileage do you get per gallon of gasoline?

(5) Do you experience any trouble from slippage in rainy or snowy weather?

(6) Have you had any serious trouble from derailments on curves?

(7) Of the different commercial designs now on the market, which do you consider the most satisfactory?

(8) Do you expect to be in the market for rail motor-car equipment in the near future and, if so, what equipment will you need?

(9) Give us your suggestions as to the necessary compartments and toilets as suggested by the demands of your service or required by your State Railroad Commission

There was a most gratifying response, indicating great interest in the subject, and from these answers we were able to arrive at certain definite conclusions.

We have a used-equipment department in the Association, which is a sort of clearing-house among our member lines as well as some of the trunk lines, and from this we secured a tabulated list of all the rail motor-cars offered for sale. This threw a most interesting spot-light on the entire subject.

We made a personal inspection of the most successful cars available and spent a great amount of time in making demonstrations and in looking over the manufacturing facilities of the firms proposing to build them. There were some cars offered that we did not examine because we considered them impractical or not soundly financed.

Summing up these three phases of our investigation, we found that the most successful cars in service were those of light design, using a thoroughly tried and proved make of motor-truck power-unit or chassis. One make in particular showed a preponderance over all others in the ratio of probably 5 to 1. We were furnished records of cars that had made as high as 300,000 miles, and been in practically continuous service for a period of 5 years. The operating cost varied from 10 to 25 cents per mile, and the gasoline consumption from 5 to 10 miles per gal.

We found the maintenance cost surprisingly low, averaging about $15 per month on these smaller-type cars and only slightly above this on the larger ones. By smaller type I mean those using a 2½ or 3-ton motor-truck chassis, and by larger cars those using 5-ton chassis. The operating cost of 10 cents per car-mile was, of course, confined to the former, which were being operated by one man. But some of the larger types using two men were being operated as low as 20 cents per car mile, as given in Table 2.

The figures in Table 2 were made on a basis of $12,500, the purchase price of the car, and an operation of 100 miles per day.

TABLE 2—COST OF RAIL MOTOR-CAR OPERATION

	Cost per Mile
Gasoline	$0.030
Labor, two men at $125 per month	0.085
Depreciation, rate 12½ per cent	0.042
Interest and Insurance	0.022
Maintenance	0.021
	$0.200

It was revealed that the majority of cars on the short lines were being operated most successfully by younger men, who were trained as mechanics and who were thus able to take care of practically all of the necessary light repairs. I think this point should be emphasized strongly, as the labor cost is one of the principal single items in the operation of rail motor-cars, and the payment of standard wages would defeat the object to be accomplished. At least this is true on the short lines. We take the position that the operation of these cars does not require the skill or training necessary to operate a steam locomotive. They have never been classified by the Interstate Commerce Commission, and I feel sure that our position would be upheld.

The consensus of opinion was that all cars should be equipped with a pivotal lead truck for safety and that a single pair of drivers with the proper weight distribution gave satisfactory service, though the riding qualities of the car were naturally not so good as if a four-wheel truck with swing bolster were employed.

The different rail motor-cars offered through our clearing-house revealed that practically every road owning the old heavy and now obsolete types, some of which are not now being built, desired to sell them at prices ranging from $500 up. Of all the cars offered, however, there was not a single one of the modern light adapted truck type.

In our personal examination and inspection of the different cars offered, we found that the light six-wheel type cars up to a length of 36 ft. and a weight of about 20,000 lb. could be operated successfully at a speed of from 30 to 35 m.p.h., making from 5 to 6 miles per gal. of gasoline. Beyond this, there is too much vibration, and the single-driving-wheel arrangement makes the car ride uncomfortably; but for a capacity up to 35 passengers and about 2000 lb. of baggage, we found this the most successful car of the present time. Above this capacity, we examined a car 43 ft. in length, equipped with two four-wheel pivotal trucks that was capable of making a maximum speed of slightly better than 40 m.p.h., at which speed it rode very comfortably. The weight of this car was about 30,000 lb.

In conclusion, I believe that these cars will prove the salvation of many short-line railroads, as well as the branch lines of the larger systems; and, as a large number of our member lines have stated, they have changed their figures from red to black. While the cars that we have been discussing are absolutely successful and will faithfully perform the duties imposed on them, I feel that efforts should be expended toward the development of a higher-powered engine, as the present ones have none to much power. I do not mean to increase the bore of the cylinders or go to the slow-speed marine-type of engine; but, instead, to increase the number of cylinders and adhere strictly to the successful and proved type of automobile engine.

THE GASOLINE-DRIVEN MOTOR-COACH FOR RAILROAD SERVICE[1]

By Charles O Guernsey[2]

Efforts to operate railroad rolling stock by gasoline engines have met with little success in comparison with the results achieved in other lines. To bring out the reasons and to show what the field is for this class of equipment is the purpose of the present paper. The various attempts to adapt gasoline engines to railroad work are reviewed from the days of the Strang and McKeen cars. To demonstrate that the real field of the gasoline railcar lies in the middle-ground between the rail bus and the steam train, the author enumerates the advantages of both kinds of service and shows that, to develop a satisfactory car, each side of the controversy must make concessions to the other. The early difficulties having been largely due to the use of too heavy cars, the weight could be decreased materially by the application of approved features of automotive practice. Other requirements should include four-wheel pivoted trucks, front and rear, full speed in either direction, air-brakes and safety appliances. The motor-coach should combine the light weight of the motor truck with the safety, steadiness, comfort and convenience of the steam coach.

Since the days of the first automobile, various attempts have been made to design railroad equipment

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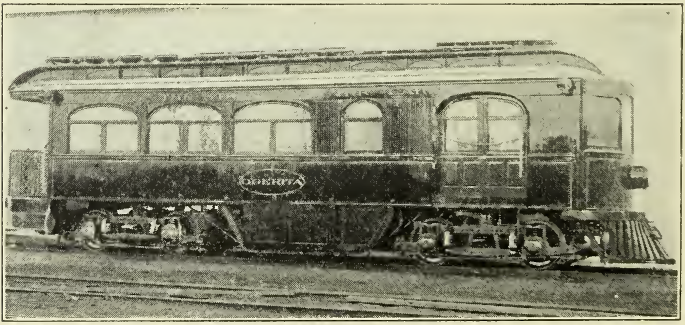

Fig. 1—An Early Motor Rail-Coach, the Strang Gas-Electric Car That Was Built in 1905

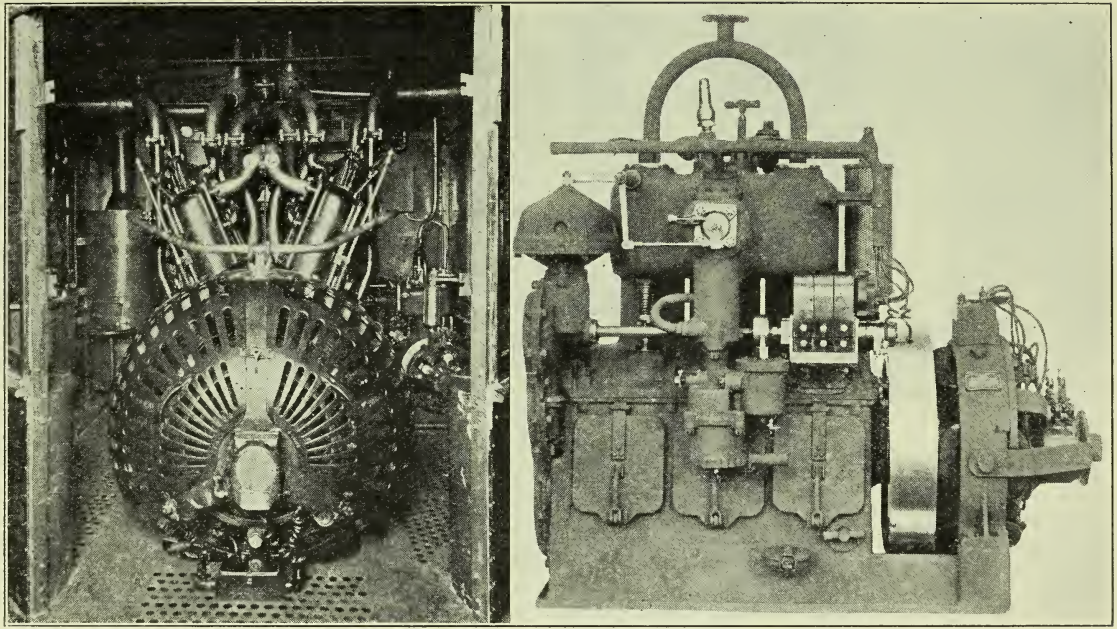

FIG. 2—END AND SIDE VIEWS OF THE GASOLINE-ENGINE GENERATING SET DEVELOPED BY THE GENERAL ELECTRIC CO. IN 1910

operated by gasoline engines. It seems rather remarkable that so little progress has been made, when one considers the place that the gasoline engine has in the marine, aeronautic, automotive, stationary and, in fact, practically every power field except the railroad. I am of the opinion that there are definite reasons why progress has been slow up to the present time. It is the purpose of this paper to bring out these reasons and to show what the present field is for this class of equipment. The motor-coach will fill a decided need but it must not be considered as a cure-all. It has limitations.

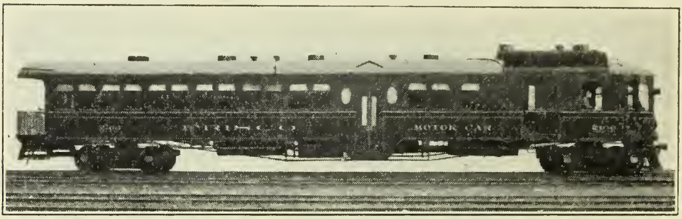

Fig. 3—The General Electric Motor Rail-Coach

One of the early attempts was that of Strang, which occurred about 1905; it is illustrated in Fig. 1. Previous to this, various efforts had been made to provide a single-unit car with steam. These cars were unsuccessful primarily on account of boiler limitations, the light fire-engine type of boiler being too expensive to maintain and the locomotive type too heavy. They had no particular advantage over the ordinary type of steam locomotive since they did not eliminate roundhouse supervision and the like. About 1910 the General Electric Co. became interested in the problem and spent considerable time and effort in developing cars to be propelled by electric motors, the current for which was to be supplied by a 175-hp. gasoline-engine, connected to an electric generator as shown in Fig. 2. Notwithstanding that a number of these cars, such as the one shown in Fig. 3, were built, and some of them, in fact, are still in service, generally speaking, they were not successful because the great weight, complication and maintenance expense made their operating cost almost as high as that of a steam train. Mr. McKeen, of the Union Pacific Railroad, seeing the need of something of this kind, developed the McKeen car, of which probably more have gone into service than any other up to the present time. Here again, the great weight necessitated an engine considerably larger than was commercially practicable, so that the total cost of operating the car was only slightly less than that of a steam train. Another reason why these early cars were not popular was that the builders failed to give due consideration to the economics of the situation. They attempted to have the same capacity and speed in a gasoline car as in a steam train consisting of a locomotive and two cars. Wherever such capacity is required, the steam train can ordinarily be operated at a profit. The real field for the gasoline car is in service where the steam train has more capacity than is required. The heavy gasoline-cars referred to above ran too near to the steam train in capacity, speed and operating cost. They did not fit the field.

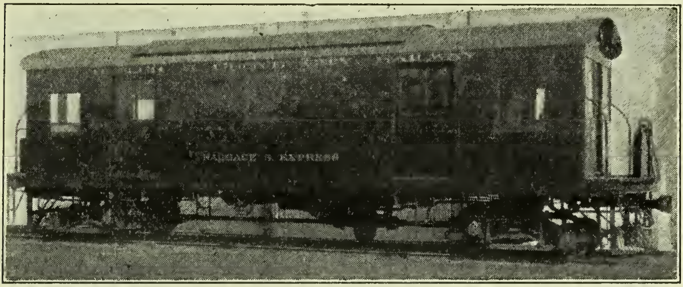

FIG. 4—ANOTHER EARLY MOTOR RAIL-COACH IN WHICH A SILENT-CHAIN DRIVE AND PROPELLER-SHAFT WERE USED

About 15 years ago the car shown in Fig. 4 was built at the plant of the J. G. Brill Co., Philadelphia, on contract for the inventor. The engine was mounted in the center of the car, being connected by a silent-chain drive to a countershaft from which the drive was through propeller-shafts to two bevel-geared axles, one on either truck, as shown in Fig. 5. Other cars have been built similarly, including those of Hall-Scott and Sargent. Cars that have failed did so because the designers failed to see the peculiar field of the gasoline engine.

It is generally conceded that for continuous heavy-duty work, engines having cylinders with a bore larger than 5 in. are not commercially successful. The troubles due to warpage, lubrication difficulties, the heating of valves and piston heads and the like become too great to handle. Apparently, the failure of these cars can be traced directly to the failure to appreciate the limitations of the gasoline engine.

On the other hand, builders of motor trucks for several years have equipped chassis ranging from ¾ to 5-tons capacity with flanged driving-wheels and with other means to adapt them to operation on rails, as shown in Fig. 6. These cars, in general, have been successful. Due to light weight, low rolling-resistance, the small engines required, and to the fact that in some installations they can be handled by one man, the operating cost has been exceedingly low. These cars, however, being unduly limited in capacity and speed, fill only a limited demand. There still remains a middle-ground between the rail motor-bus or the rail motor-car, as we have known them in the past, and the proper field of the steam train. It is in this middle-ground that the motor-coach can make a place for itself.

A motor-coach is defined as a passenger-carrying gaso-

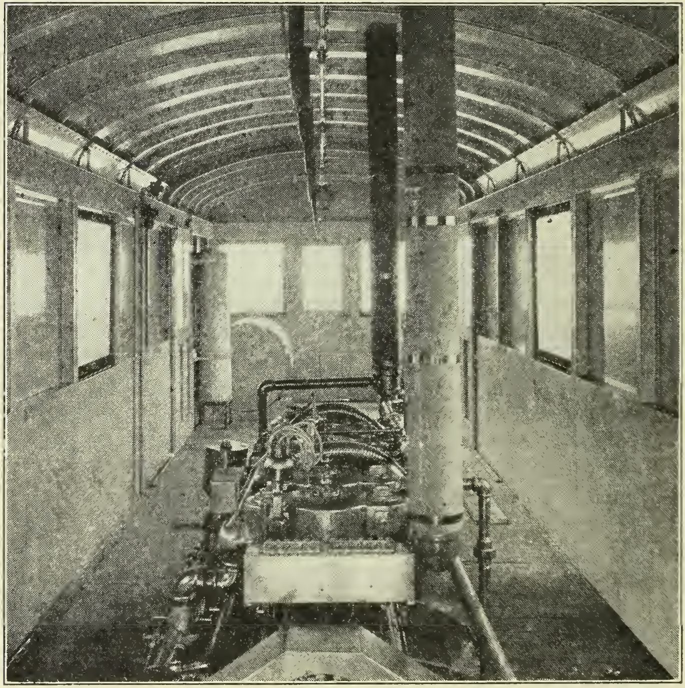

FIG. 5—POWERPLANT OF THE CAR ILLUSTRATED IN FIG. 4

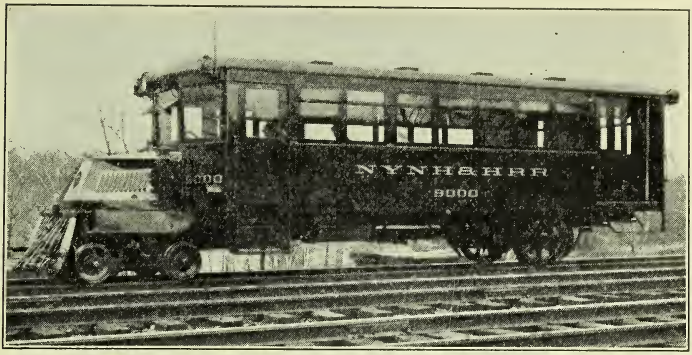

FIG. 6—A MOTOR-TRUCK CHASSIS EQUIPPED WITH FLANGED DRIVING-WHEELS FOR OPERATION ON RAILS

line-driven railroad motor-car, designed specifically for operation on rails. It is in no sense a converted motor-truck, but represents a combination of automotive and railroad practice. Previous attempts have been based almost entirely on one to the exclusion of the other. To illustrate, some of the earlier cars developed by the railroad men weighed more than 50 tons and required 300 hp., although seating capacity was provided for only some 50 to 60 passengers. They did not give due consideration to the automotive side of the design. Instead of building a gasoline car, they attempted to use a gasoline engine in a car of steam-train design.

By careful design and the use of alloy steel, anti-friction bearings and other approved features of automotive practice, the weight can be reduced materially. The weight must be held to a minimum to keep the motor-coach requirements within the capacity of proved gasoline-engines. It is also undesirable to forget the railroad point of view entirely. Many features of railroad design are the result of almost a century of development. The designer must weigh his problem carefully, choosing from railroad practice those features that fit this new type of equipment. The converted motor-truck does not, according to experienced railroad officials, meet the requirements. Due to the use of two-wheel driving-trucks and other practices, it does not ride as steadily or as safely as the usual railroad car, which has a four-wheel truck under either end of the car.

There is an insistent demand for several features not ordinarily included. Some of the more important of

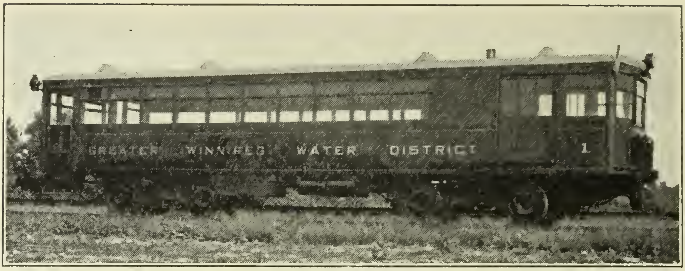

FIG. 7—A 44-FT. MOTOR-COACH HAVING A TOTAL SEATING CAPACITY OF 46 PASSENGERS

these are four-wheel pivoted trucks, front and rear, full speed in either direction, air-brakes and safety appliances. The motor-coach should combine the light weight of the motor truck with the safety, steadiness, comfort and convenience of the steam coach. Fig. 7 illustrates a motor-coach, having an overall length of about 44 ft. and a seating capacity of 38, in addition to drop-seats for 8 passengers in the baggage-room, making a total seating capacity of 46, as shown in Fig. 8. The baggage space is 70 sq. ft. The car is provided with standard vestibule-doors for entrance, a saloon, comfortable seats, electric lights and other features commonly associated with modern railroad design.

The total weight of this car is only 13 tons. This reduction of weight to less than one-third that of old-time motor-cars of the same capacity makes it possible to use a 68-hp. engine, as against the 200-hp. engine required by other types. At the same time a speed of 48 m.p.h. has been attained and a speed of 35 m.p.h. can be maintained indefinitely, without damage to the mechanism. Due to the light weight and the correspondingly small amount of power required, a car of this type will show exceptional economy. The fuel-consumption is light, the car running between 5 and 7 miles per gal. of gasoline. Due, also, to the light weight, the car has very good acceleration, reaching a speed of 25 m.p.h. in 30 sec. from a standing start. This car is arranged with two four-wheel pivoted trucks. The drive is from the unit powerplant, located forward, through an auxiliary transmission, contained in the bolster of the front truck, shown in Fig. 9, to the two axles of the front truck. The auxiliary transmission is arranged so that either of two pairs of gears can be used for transmitting the drive,

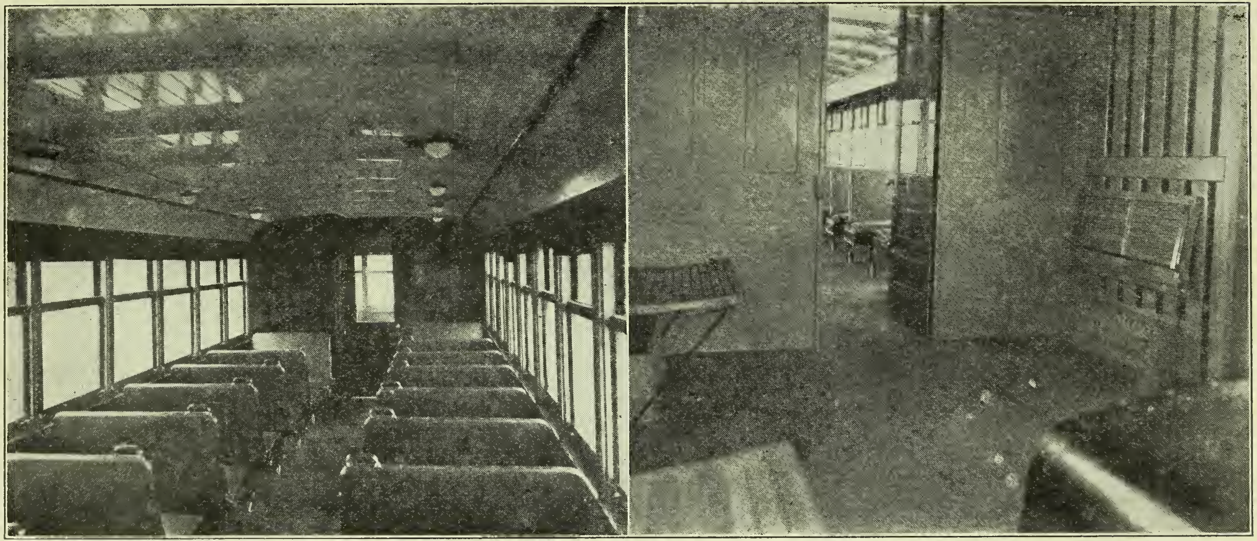

FIG. 8—VIEWS OF THE PASSENGER AND BAGGAGE COMPARTMENTS OF THE CAR SHOWN IN FIG. 7

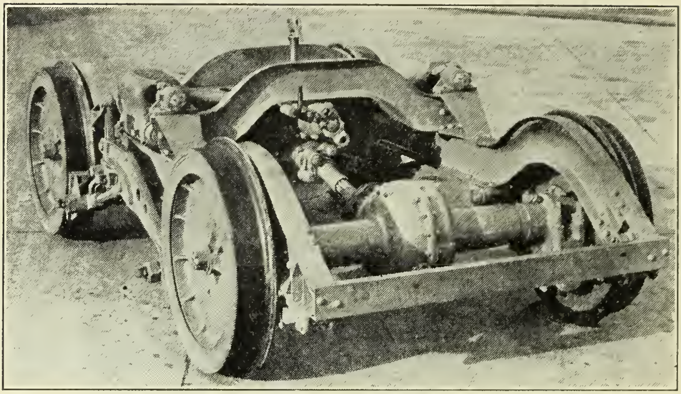

Fig. 9—Front Truck of the Car Illustrated in Fig. 7 in Which Power Is Delivered to Both Axles by an Auxiliary Transmission Contained in the Truck Bolster

thus, in effect, giving two high-gears. One of these gears is proportioned for the ruling grade on the particular railroad on which the car is to be used, and the other is proportioned to give a maximum speed in straightaway operation.

Service Possibilities

The success of the motor-coach, after all, hinges primarily on the engine. The car must be designed with this thought always uppermost. The engine must be one that will stand up under the severest service. It must be capable of operating continually at high speed and under wide-open throttle with the minimum of vibra-

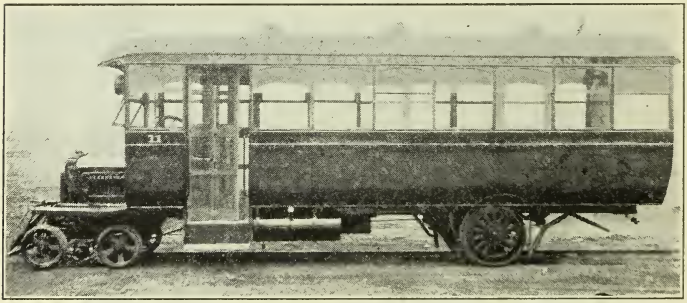

Fig. 10—A 30-Passenger Gasoline Motor-Coach That Is in Service on a Branch Line in the Blue Ridge Mountains

tion. Everything must be accessible and so arranged that repairs can be made quickly.

The gasoline-driven railroad motor-coach will enable many branch lines and short lines that are now operating at an enormous loss to be converted to a money-making basis on account of the low cost of operation, maintenance and the like. A 13-ton car can be operated for 25 to 35 cents per car-mile, as against a cost of $1 to $2 for steam operation. The initial cost is low.

Frequent service could be given where it is not justified now by a steam train. A freight-carrying unit might be installed in conjunction with a passenger unit at an initial cost much below that of a passenger unit. The gasoline motor-coach as a unit possesses many advantages over steam and electrified service; namely, speed, frequent service, low cost of operation, a crew of one or two men, the elimination of the usual terminal facilities and a great reduction in the initial cost. In conclusion, the railroad motor-coach is primarily designed for service where steam operation is too expensive and frequent service is desired.

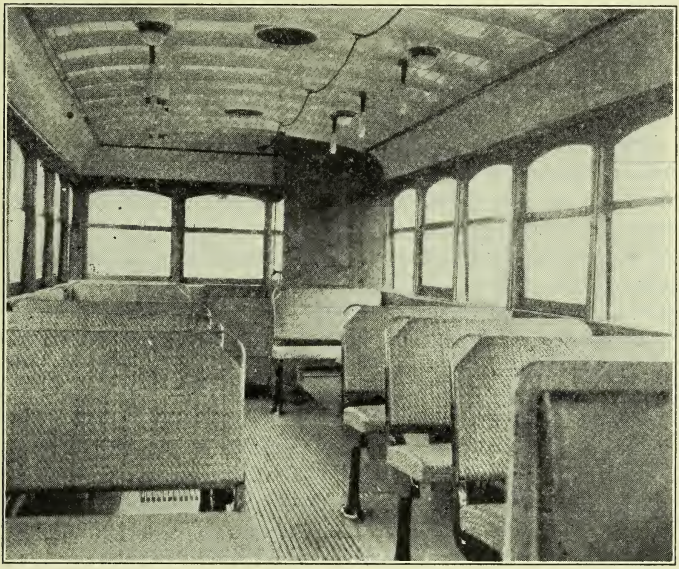

FIG. 11—INTERIOR OF THE CAR SHOWN IN FIG. 10

There is a small branch-line in the Blue Ridge Mountains that was getting only about $30 per day in passenger revenue while the operating cost with a steam train was $46 per day. A 30-passenger gasoline motor-coach, illustrated in Figs. 10 and 11, was installed and is running for about $10 per day or about 11 cents per mile; this is the total operating cost. Coal docks, water tanks, cinder pits, hostlers and the like are not necessary. That branch line, running from "nowhere to nowhere," is now making $20 per day profit. This sort of operation is a live issue with the railroads. As a result of this showing this road has put on a second car and is doubling its service voluntarily because it pays. The larger cars will vary in operating cost according to their capacity and speed. A car of this type can be operated for 20 to 30 cents per mile, depending upon the track, speed, number of stops and other conditions.

THE DISCUSSION

C. CHANDLER:—Mr. Hall, vice-president and general manager of the New Orleans & Lower Coast Railroad, states that the operation of its gasoline railroad motor-car is costing 15 cents per mile, without depreciation. I understand that a trailer, to be installed in the near future, has been ordered. The Illinois Central Railroad is, of course, very much interested in anything that will reduce operating expenses. With this idea in mind the management has started an investigation covering the use of motor cars on branch lines. At the present time we do not know just what is desired in the way of a railroad motor-car. On account of the race question in the Southern territory, it is possible that a car which would be satisfactory on the Northern lines would not answer at all in the Southern districts. The investigation so far indicates that the handling of freight by motor car is not desirable.

CHARLES O. GUERNSEY:—I think that about 75 hp. would be the outside limit of engine capacity. Where the roads are fairly flat, it is possible to haul a light trailer behind one of the cars. Almost every case has to be considered alone. I would say that when hauling passengers and no express on an ordinary road having less than 1-per cent grades, from 50 to 70 people could be handled. The time schedule is a governing factor.

B. M. FREDENBERG:—I notice that in riding there seems to be a perpetual jar. The engine operation jars everything loose. The windows rattle and you think there is a hail-storm. It must be due to the constant rattling of the engine. I am wondering if that has been overcome since I rode on a gasoline coach, operated by the Ann Arbor Railroad and the Pere Marquette Railway several years ago.

MR. GUERNSEY:—That car on the Ann Arbor line weighs 54 tons. This means that they must have about 200 hp.; that they must go beyond the commercial limits of the gasoline engine to get sufficient power; and that the big heavy engine vibrates to such an extent that it becomes uncomfortable for passengers. The thing we need to do is to make the car light enough to conform to the size of engine that has been proved. With a light car it is possible to get better acceleration with 60 hp. available than they got with 200 hp. Another point is that those heavy cars, because of the enormous weight and power required to move them, cost almost as much to operate as a steam train. The operating cost varies. We get a different report everywhere we go. Conditions vary, of course.

MR. CHANDLER:—They are operating at a cost of about 40 cents per mile, with a passenger revenue of about 55 cents per mile, not including revenue from milk, express or mail service. That is the only car of this kind that I know of.

L. G. PLANT:—Light local-passenger service, which involves the operation of a steam locomotive and not more than three coaches, is undoubtedly the most expensive service performed by the railroads in proportion to the revenue received. Testifying before the Interstate Commerce Commission in regard to railroad problems and the efficiency of railroad management, Mr. Willard, president of the Baltimore & Ohio Railroad, recently made this significant statement regarding railroad passenger service:

> The expensive feature of the passenger business is not the ordinary running of the heavy through-trains. My observation is that usually they pay. The expensive part of the business is the running of thousands of miles of unprofitable passenger service on light branches, or light portions of main lines where the people demand the service. I have not made figures recently, but I recall that a few years ago some 30 per cent of the total passenger traffic carried by the Baltimore & Ohio Railroad earned less than the actual out-of-pocket cost. We earned less money than the actual wages paid to the trainmen and enginemen, for the coal

> burned and the oil used, and there was not a single train either that we could take off. They were established and have been run in response to the public demand; and I am not in position to say that the public is not entitled to the service. Usually the service consisted of two passenger trains in each direction over branch lines. The business did not justify running the trains, but the people had no other way to travel.

While Mr. Willard did not refer to the steps taken by his railroad toward reducing the cost of this service, it is understood that the Baltimore & Ohio Railroad is actively interested in the possibilities of the gasoline motor-car. *The Railway Review* also is interested in the possibilities of the gasoline rail motor-car and from the time the first article describing this class of equipment appeared in this paper some months ago, no one subject has created greater interest. It is our belief that in the present and prospective designs we have the solution to the problem outlined by Mr. Willard. In fact, my experience some years ago in the purchase of a small railroad in a mountainous section of Virginia convinced me of the value of the gasoline rail motor-car as a means for handling light local-passenger traffic. This road had been a losing venture for many years and, in planning for the future operation of the road, I conceived the idea of purchasing a motorbus, such as I had seen operating on the streets of Birmingham, one of the first Southern cities to adopt bus transportation extensively, and equipping it with flanged wheels. I have since been a firm believer in the gasoline railroad-car as a practical means of reducing the cost and improving the character of local-passenger service, provided the elements that have contributed to the success of the motor truck, such as simplicity of operation and maintenance, light weight and low first cost, are not ignored. It is interesting to observe that a recent investigation into the use of gasoline railcars on short-line railroads revealed the fact that in every instance the substitution of gasoline service for the steam locomotive had converted an annual deficit into a surplus, although in many instances gasoline equipment of the most primitive type was being used.

In view of the interest on the part of railroad officials, it is perhaps puzzling to the builders of gasoline cars to find so much hesitancy on the part of the railroads toward the actual purchase of the equipment, even where the capacity of the railcar has been demonstrated and the operating economies are obvious. If a motor-truck builder can convince a local coal-dealer that he can save $1,000 per year through the substitution of a motor truck for team delivery and the dealer can finance the purchase, the sale of the truck is ordinarily assured, but it should be borne in mind that in this case the profit from the investment will accrue directly to the dealer, whereas the meager profit resulting from railroad operation is paid to stockholders whose connection with the actual details of the service is usually remote. I do not wish to indicate that railroad managers and employes are not making a sincere effort to improve the efficiency of railroad operation in every possible way, but their less responsive attitude can be ascribed chiefly to a lack of business initiative that is more or less characteristic of all large institutions.

While railroad officials are undoubtedly interested in the development of the gasoline railcar, they are inclined to see the objections to this equipment rather than its possibilities and to stress the importance of certain changes in the details of the construction of the cars instead of the lower operating cost of and small investment required for the equipment in its present form. Railroad officials also are inclined to want equipment that will duplicate the existing steam service. In the situation described by Mr. Willard, the question is not necessarily that of duplicating the performance of the two passenger-trains that operate daily in each direction over branch lines, but of expanding the service and of building up additional traffic at less cost than is possible with steam operation. It is conceivable that on local service radiating from a shopping center a gasoline railcar could be maintained in continuous operation throughout the day and handle a greater number of passengers at less cost than is possible with a steam train. It is reasonable to assume that more frequent service would stimulate a greater volume of traffic. I have in mind a typical situation on a short branch-line where there is but one train into town early in the morning and one train returning late in the evening. The installation of a frequent gasoline railcar service throughout the day would not only attract more travel, as in the case of the electric line, but enable the railroad to handle a greater number of passengers per day at a lower cost. Notwithstanding these obvious possibilities, we find that one railroad is not interested in gasoline railcars unless the cars are designed to seat 72 people, while another road insists that the equipment must be capable of pulling four or five box-cars if necessary. Still another railroad expects to use its old coaches weighing some 60,000 lb. as trailers rather than charge this equipment off its books, this being a case where the railroad has failed to make a sufficient depreciation allowance for the coaches and is carrying them at a higher book-value than they are actually worth.

On the other hand, there are certain requirements of the railroad, particularly those that relate to safety, the importance of which may not be fully appreciated by manufacturers, but which designers, in their desire to make the equipment as light as possible, should not ignore. Builders of gasoline railcar equipment cannot yet be said to have a full understanding of the problems involved and an appreciation of the fact that it is not what the car is, but what it will do, that interests the practical railroad man. Generally speaking, the manufacturers have not differentiated between the problem involved in marketing trucks and the problem of selling to the railroads. It will necessitate an organization that understands the railroad problem and the full significance of departmental relationship that is foreign to the ordinary commercial organization. I believe that the arrangement to market gasoline railcars through local agencies that receive a large commission would not appeal to the railroads, and that the service feature so essential to the truck business would not meet with favor on the average railroad. It will be necessary in a majority of cases not only to demonstrate that the car will operate, but to show the railroads where they can use the cars to advantage; and this involves a large amount of educational work.

CHAIRMAN LON R. SMITH:—Nobody would claim that the gasoline railroad-car would take care of the service that demands a steam outfit. The branch-line service, which is necessary on a steam line, makes the advantages of these gasoline-propelled cars of interest. Most of the railroad people seemingly are interested in the subject. The question is, what type of car will best meet their needs.

J. D. RISTINE:—The railroad motor-coach is of vital interest. It can be used successfully where the cost of steam operation is too great and frequent service is required, at a cost approximately one-fifth to one-third that of a steam train. The gasoline engine is, first of all, the all-important thing to be considered. We must not fail, however, to consider its limitations and construct a coach accordingly. For the present we are limited to engines of about 75 hp. as a maximum, for the reason that larger engines have not been perfected to the point where they are commercially successful. Coaches should be designed so that minor changes can be made to meet conditions, but in all cases the total weight must be given careful consideration.

MARK A. SMITH:—The car that Mr. Guernsey has described was driven from Wabash, Ind., today, carrying a group of railroad men as passengers. The running time from Peru, 75 miles distant, was 2 hr. and 35 min. The time on the steam train would have been 13 min. longer. The maximum speed attained in this run was 48 m.p.h. The coach took the grades at from 25 to 30 m.p.h. The gasoline consumption from Wabash, 91 miles, was 15 gal., including idling; or 6 miles plus per gal.

A. L. NELSON:—That is very good economy.

ALBERT KING:—I am a road foreman of engines on the Wabash Railroad. The car mentioned is a very simply constructed affair. I ran it myself the first time I was ever on it. I own an automobile but I do not consider myself an expert driver. The car has many features that could be used to advantage by railroads. I take it that they all have the same conditions that we have. A steam locomotive never operates at its maximum efficiency except at or near its maximum capacity. If it had sufficient tractive power to haul 2000 tons over a 100-mile division and the coal consumption were, say, 6 tons, under the best and most economical management it would not take one-half that load over the same division with one-half that amount of fuel. That is why the operation of branch lines is so expensive. The only thing that I could suggest to Mr. Guernsey and his fellow-workers is to perfect a car that will meet all the requirements of branch-line or local-passenger service. The cars that are being designed are a little too small. A 35, 40 or 50-passenger car is hardly large enough, and the trailer feature is not, to my mind, desirable. The men engaged in building railroad motor-cars have a broad field.

GEORGE A. WEIDELY:—Why did Mr. Guernsey specify the limit at 75 hp.? Engines used in the fire-pump service develop 100 to 120 hp. continuously and economically under severe conditions, making runs of 30 to 50 hr. It seems to me that engines of that kind would be very satisfactory in railroad service. I am also wondering what Mr. Guernsey's idea is as to the number of cylinders most suitable for that service.

MR. GUERNSEY:—I may have been a little unfortunate in the manner in which I stated the horsepower limit. We do not say that we have the ultimate thing. I see no reason why ultimately it should not be possible to have a larger car with more power but with an increased number of cylinders. Of course, we must remember that increasing the capacity of the car increases the fuel cost and the operating cost. It is in the haul in which you get 40 people sometimes, but usually 6 or 8, where these things really belong. I think there are great possibilities for the future development of these cars.

IRA C. KOEHNE:—A few weeks ago I took the train from New Haven toward Waterbury and rode about 40 min. on a gasoline-driven railcar. That ride was a punishment on account of the excessively short jerky vibrations, not only from the engine, which is in the front of the car, but from the rear axle as well. I think it would be a menace to the health of passengers to ride on a car of that construction. I can see that the construction pointed out by Mr. Guernsey has features that cushion the vibrations. Economy of operation is a very desirable consideration, but to make the thing a success the comfort of the passengers must be conserved.

E. O. MANSUR:—I have had the benefit of all the grief that came with the introduction of the gasoline rail motor-cars, having operated them since 1914. I am operating them on the Akron, Canton & Youngstown Railway at present.

The gasoline-electric motor-car with which I am most familiar has an eight-cylinder engine, 8-in. bore and 10-in. stroke; it runs at 550 r.p.m. and gives 175 hp. This engine is direct-connected to a 600-volt series-wound generator that furnishes the current for two 100-hp. motors on the front truck. The most trouble was encountered because the steam engineers did not understand the gasoline engine and did not know what to do in emergencies. A short time ago an engineer called me up from the road, reporting that a valve had broken and dropped into the cylinder. He wanted to know what to do. I told him to shut down the engine, but it was too late. The damage was done. These engines have too many moving parts and are too complicated for an inexperienced man to watch.

The electrical equipment has, and has not, given trouble, depending upon where the car is taken care of. In some cases the men do not realize the necessity of keeping the oil away from the motors, and I know of one case where an armature was saturated with oil. When the oil is kept out of the motors, no trouble occurs. I am of the opinion that these cars could still be made to operate efficiently if an automotive engineer would study them and make a few changes in ignition and carburetion.

While these cars weigh 35 tons and 60 per cent of this weight is on the front truck, I believe we can still go ahead with the large type of car and reduce some of the weight by using a high-speed engine. With the past types of construction, the railroad men have condemned the gasoline engine for railroad service because of the failures and delays on the road that are due to the breaking of small parts and to having them operated by inexperienced men.

W. H. BUDERUS:—I am interested in lubrication. Our greatest trouble in the earlier cars was on account of the oil in the crankcase becoming diluted so quickly. Those crankcases held 26 gal. The lighter type of engine, which Mr. Guernsey has explained, holds from 2 to 3 gal., a tremendous saving. In regard to the engine of lower horsepower, I happen to know that a car is now under construction which will use a 100-hp. engine. I understand the Baltimore & Ohio Railroad estimates that it could place 50 gasoline railcars on its system alone; that is, there is an opportunity for this number of cars to replace unprofitable steam-train service.

MR. GUERNSEY:—It is well within the possibilities to build a car for way-freight service and for use where it might be required to handle one or two loaded cars. In fact, the operation of these cars can be pretty well illustrated by what is being done on the electric interurban lines. They haul way freight in carload lots and by a trailer. These cases come up on branch lines, where the speed need not be more than 20 or 30 m.p.h. A car-and-trailer arrangement can be worked out, provided the speed conditions and grades are not too severe. We run about 700 miles per gal. of oil. As to the noise from the engine, and the vibration and rough riding due to the reaction from the driving axle being objectionable to the passengers, these are things we have set out to overcome. They are among the reasons we are using two four-wheel trucks. The point was raised regarding bus competition.

A car running on the rails can starve a bus man to death. A 30-passenger bus operating on the highway costs about 30 cents per mile to operate, while one running on rails will cost about 12 cents per mile. These cars, because of their light weight, require very little expenditure for maintenance.

EXPERIENCE NOTES FROM A PRODUCTION NOTEBOOK[1]

By H J Crain[2] and J Brodie[2]

While investigating the sources and causes of noise in automobiles during an extensive connection with one of the largest automobile companies, the authors recorded their experiences in the shop in the form of notes. Some of these are offered with a view to stimulating the discussion of the subject and with the hope that additional information will be brought out by an exchange of ideas, particularly on the problem of eliminating gear-noises. In many cases they found that noise was caused by failure to allow sufficient clearance for an adequate oil-film. And it was noted frequently that when one noise had been located and silenced another appeared that had not been apparent before. The topics that have been considered include the running-in of brake-bands, engine knocks, oil-pump gear-noise and that of gears in general, the clearances of ball bearings, backlash, and rear-axle bevel-gears.

The experiences recorded in this paper have not been selected in accordance with a specific plan. No attempt has been made to cover any particular subject fully or to arrange the different descriptions with regard to a related sequence. It is possible that some of the experiences have been encountered or the methods have been used by other production executives but we believe that knowledge of these methods is not general and that it will be of interest to factory men. We hope that the discussion will bring out additional information on some of the matters treated, particularly on the perplexing problem of eliminating gear noise.

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[1] Detroit Production Meeting paper.

[2] Production department, Packard Motor Car Co., Detroit.

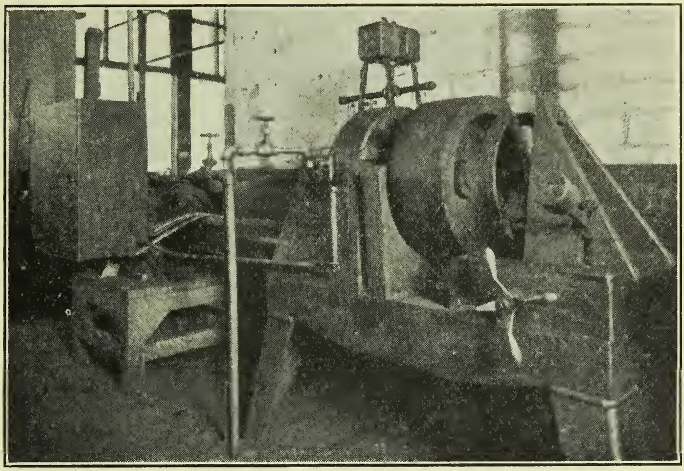


FIG. 1—MOTOR-DRIVEN MACHINE FOR RUNNING-IN INTERNAL BRAKE-BAND ASSEMBLIES

RUNNING-IN BRAKE-BANDS

The increasing congestion on city streets and the seriousness of the automobile accident and collision situation should be convincing evidence of the need of proper adjustment of motor-car brakes. It would seem important that cars should be shipped from the factories with the brakes seated, and adjusted to overcome the rapid wear that usually occurs in driving the first few hundred miles. This rapid wear is caused by the ironing or smoothing of the brake-lining surface until the high spots have been worn down to the level of the rest of the lining face. It may be due also to slight imperfections in the contour of the brake-band. Figs. 1 and 2 illustrate two motor-driven machines designed and built by the Packard Company for the purpose of running-in brake-bands. The machine shown in Fig. 1 handles the internal or expanding brake and that in Fig. 2 the external or contracting brake. The brake-drums rotate at a speed of approximately 1000 r.p.m. in both cases. The drums are cooled by water, circulated about the peripheries of the drums, so that the temperatures are never excessive. Pressure is exerted on the brake-bands by a weighted lever, which can be seen clearly in Fig. 1, the weight being adjusted so that the pressure is only great enough to assure a full bearing of the band on the drum.

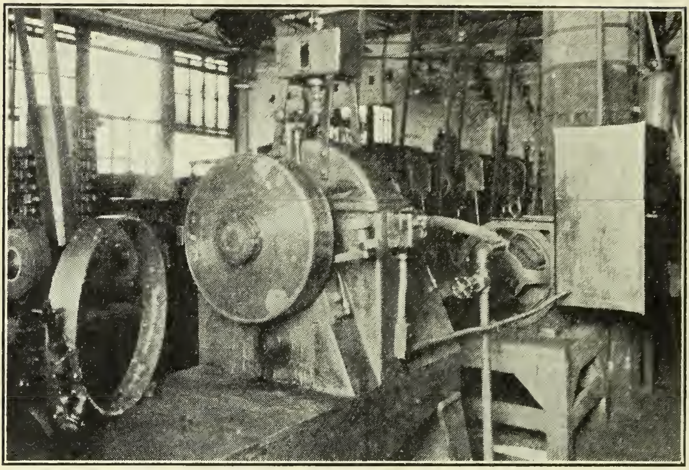


Fig. 2—A Special Machine Developed by the Packard Company for Running-In External Brake-Band Assemblies

Each band is run for about 1 min. The two machines are located so that the same operator can handle both; the band of one being run-in while the operator is loading the other. It will be found that the bands acquire a polished surface on these machines, and that the irregularities sometimes existing around the rivet-holes and throughout the lining surface are smoothed-out. By taking this precaution at the factory the maximum brake efficiency is attained at the beginning of operation of the car, and the adjustments usually required in a new car after a few days of service are unnecessary.

The importance of accuracy in grinding a piston skirt is recognized by all production men. The center, shown

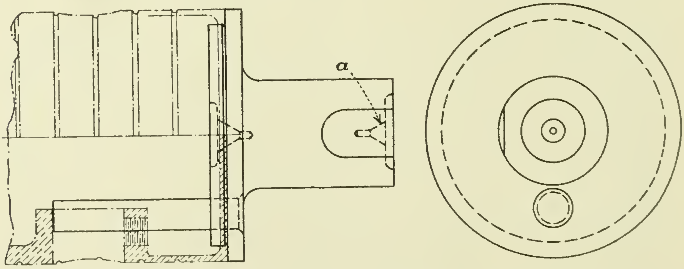


Fig. 3—Form of Center Originally Used in the Packard Shops for Centering and Driving the Pistons during the External Grinding Operation

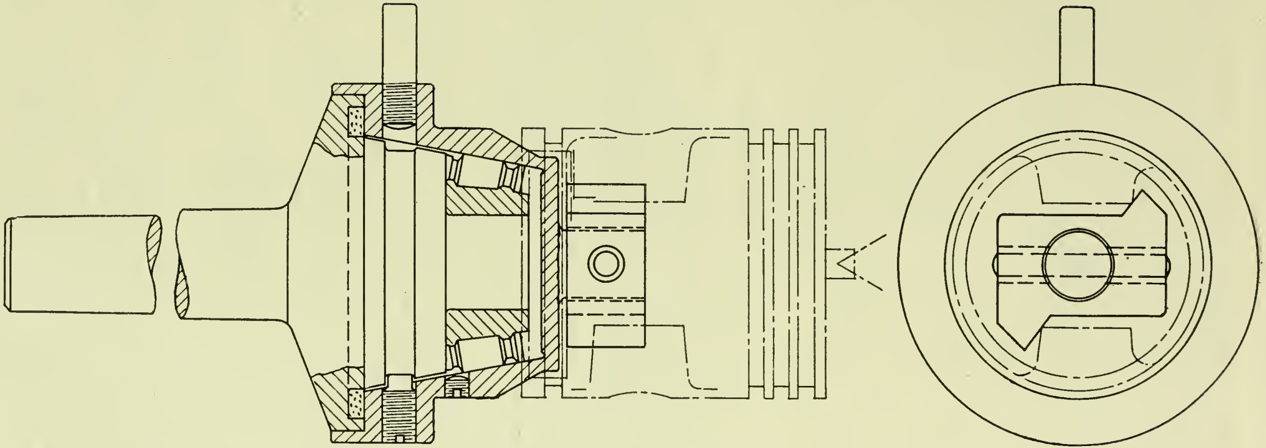


FIG. 4—A NEWER FORM OF CENTER IN WHICH THE THRUST AND THE DRIVING LOAD ARE TAKEN BY A TAPER ROLLER BEARING

in Fig. 3, was originally used in the Packard shops for centering and driving the pistons during the external grinding operation. Excessive wear of the surface *a* necessitated the frequent replacement of this center and demanded constant supervision by the foreman in order that the work should not be spoiled by continuing the use of a center that had passed the permissible stage of wear. The study given to this small but puzzling problem has resulted in the adoption of the center shown in Fig. 4. In this instance the thrust and driving load are taken by a taper-roller bearing of heavy load-capacity, the wear is distributed over a very large surface, lubrication is easily maintained and the life of the center is greatly prolonged. This design has proved very successful, and, no doubt, other tool designers could apply it to advantage. This particular center is used with Brown & Sharpe Nos. 12 and 14 external grinding-machines.

A Puzzling Engine Knock

All production and inspection departments have had the displeasure of running down peculiar engine knocks. The following note from Packard experience may shed some light on this trouble. A few years ago when a new model was started through the Packard shops the engines of the first run received at the test-stands were all found to have a perceptible piston slap or knock. Numerous remedies were tried but eventually the real cause of the noise was found largely through accident.

The click came only at the time of the explosion. Investigation revealed the fact that the valves were not centering properly in the conical surface in the cylinder. The condition is shown in exaggerated form in Fig. 5. It was found that the tool used to form the valve-seat was centered by a spindle inserted in the valve-stem guide. This spindle was too much undersize and allowed the tool to float just enough to throw the conical seat out of alignment with the valve-stem guide. As a result, the valve-spring would not bring the valve fully into the seat; the valve would hang on one side of the valve-seat until the explosion snapped the valve into the seat with a very noticeable click. Of course this part of the noise was then obviated without difficulty.

But this correction did not stop all the noise. A more annoying knock was eventually found to come from the piston-rings, which were of the diagonal-cut type with a slight clearance between the ends. As the explosion-

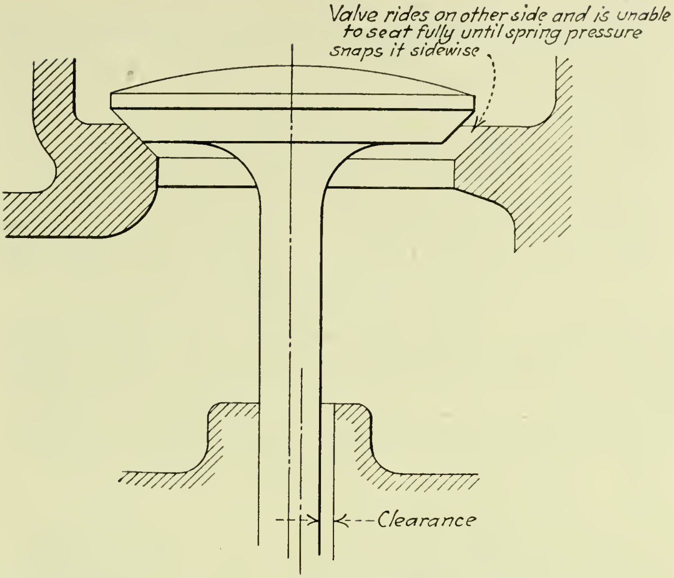

Valve rides on other side and is unable to seat fully until spring pressure snaps it sidewise
Clearance

Fig. 5—Sketch, Somewhat Exaggerated, To Show the Improper Centering of the Valves in the Conical Surface in the Cylinder

pressure reached the rings they were compressed and their ends snapped. Filing several grooves in the upper edge of the top ring would let part of the explosion in behind the ring, expand the ring and overcome the slap until the grooves had filled with carbon. A ring that slaps in the manner described generally shows bright polished ends. When the proper end-clearance had been determined by experiment, the noise ceased. The rings in general use today have overlapping joints; the end clearance can be very large and, of course, this trouble is not encountered.

Oil-Pump Gear Noise

Oil is circulated in Packard engines by a gear-pump similar to that illustrated in Fig. 6. When this particular design was first adopted it was found to produce a very irritating noise, which sounded like the blades of a fan striking a sheet of paper. Naturally this was attributed to imperfections in the gears. In the experiments made to abate this nuisance tooth-forms and pressure-angles were varied, and helical and herringbone gears were fitted, but the clatter persisted. All degrees of backlash were tried but without avail. It was noticed that a certain run of pumps were more quiet than the others. These were inspected carefully to ascertain what variation was responsible for the lessening of the noise. The only difference found was a slight relief on the lower face of the idler gear. At *a*, in Fig. 6, is shown a feeder channel that is cut in the base of the pump for the purpose of carrying oil to the idler bearing through the cross channel *b*. Note that this channel is open to the pressure side of the pump but ends at the point where tooth contact ceases on the suction side. It was found that when the tooth corners were beveled, as shown at *c*, the noise was reduced. The possible effect of these changes was the basis of a careful study, which resulted in the discovery of the real source of the noise. Both these schemes eliminated the sharp cutting off of the oil stream that would naturally attempt to escape at

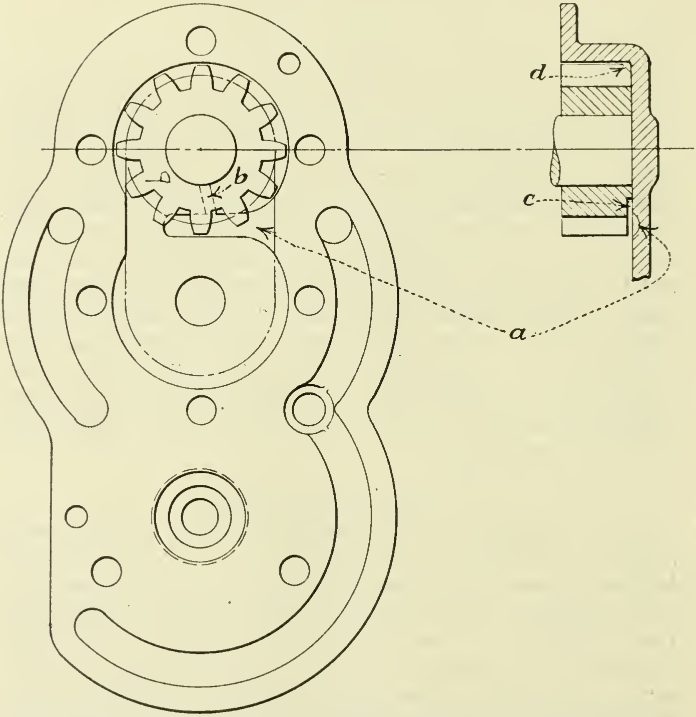


Fig. 6—The Gear-Pump That Circulates the Lubricating Oil in the Packard Engine

d from the pressure side to the suction side of the pump. By relieving the pressure in the groove *a* the oil was not able to spurt against the tooth faces and rattle the unloaded idler gear in the backlash space in the driving gear. When, as an experiment, the groove *a* was filled with solder, the altered pump became quiet. The design of the pump was changed, the groove *a* was omitted, and no further trouble was experienced. This case is cited as an example because it indicates that gear noises are not always attributable to the gear-teeth themselves.

Gear-Noise Investigations

The production and engineering staffs of the Packard Motor Car Co. have been studying the matter of gear noises for many years. This work is still being carried on but no panacea has been definitely discovered for gear troubles. Each case seems to have its peculiarities and to require special modifications to become silent. Alterations that are effective in one case might not be effective in another that to all appearances is similar. It is more than likely that the study being given to gear-tooth wear and noise in our own and other industries will lead eventually to a better understanding of the fundamental causes of the difficulties. For the present, however, it is only possible for us to exchange experiences for the common benefit of those who are interested.

A large number of investigations made over a period of years have led us to believe that gear noise does not always originate in variations of the gears themselves. Such variations undoubtedly contribute to the gear growl or chatter but the noise can often be cured or dampened by an alteration of another part. We have concluded that the proper mounting of the gears on rigid shafts is a paramount requirement if noise is to be avoided. The most perfect tooth-forms, ground, shaped or milled, will not run quietly if they are carried on shafts that spring or are not in perfect alignment. It is assumed that this essential fact is recognized in the engineer's design of transmissions and axles. Given a properly designed mounting, it is the factory man's problem to reduce noise. The factors controlling it are largely independent of blueprints. Drawings can only give the characteristics of gear-teeth; the production man must see that the actual contours and allowable variations keep them within an acceptable range of quietness.

The front end of a typical transmission is shown in

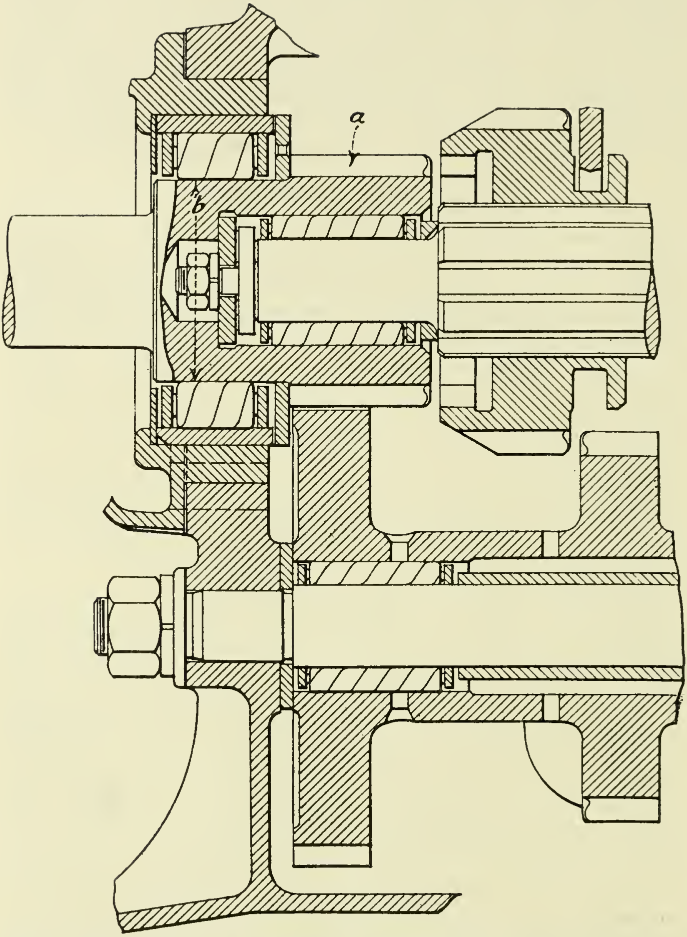


Fig. 7—Front End of a Typical Transmission Showing How the Main Drive Gear Is Carried on a Roller Bearing

Fig. 7. Particular attention is directed to the mounting of the main drive-gear *a,* in which it will be seen that this is carried on a roller bearing, the inner race of which is formed by the shaft itself and the outer race is mounted in the transmission case. In the final inspection of a certain model at the Packard factory it was noticed that the degree of gear noise varied from very quiet to objectionably loud. Attention, naturally, was centered on the noisy gears. These were returned to the transmission department and torn down for careful inspection, adjustment and reassembling. Invariably the inspection revealed gears, bearings and shafts that were as near perfection as it seemed possible to approach. This led to the assembling of special gears in which perfection was carried to the utmost degree, a state far beyond that possible under even unreasonable inspection practice. But the noise, if anything, was worse.

It remained for us to tear down and to inspect several transmissions that were passed in the final car-test as being quiet. When this was done it was found that the roller race on the shafts had been ground to the low-limit diameter, while the roller race in the shells had invariably been ground to the largest or extreme high limit specified for these holes. For purposes of comparison, six noisy transmissions were then torn down, the bearing diameter *b* was ground approximately 0.001 in. under the former low limit and the transmissions were reassembled and tested. This change caused the noise practically to disappear.

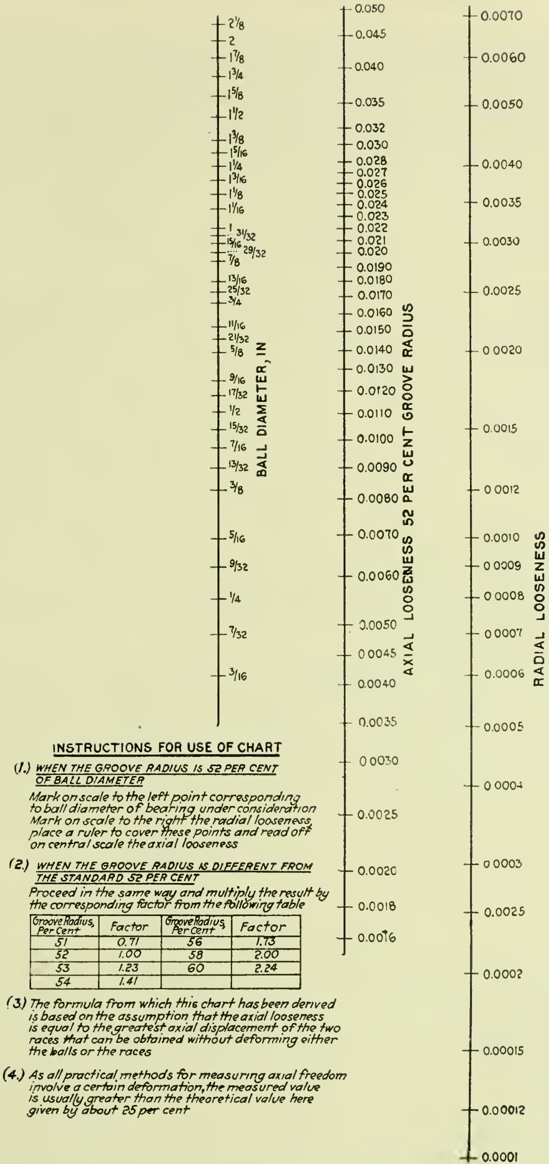
Experiments were made repeatedly with noisy transmissions and, in every case, when this alteration was made and the bearing clearance was increased, the noise was either entirely eliminated or was reduced to a degree that was not objectionable. We concluded that this result was produced by providing sufficient space for an adequate oil-film. Further experiments along similar lines have substantiated this conclusion. This explanation seemed logical since we have always found such a clearance to be necessary in crankshaft and connecting-rod bearings.

The remedy seemed a simple one to supply but we were quite concerned about mounting a bearing under conditions that simulated those it would assume after several months' wear. We had always supposed that bearings of the anti-friction type must be mounted snugly. Before definitely adopting the new practice, wisdom demanded that we check the effect of the greater diametral clearance on the wear of the bearing. Transmissions were run under similar conditions with the standard or snug bearing and with the increased clearance. We found that the snug bearing wore rapidly during the early stages of the test and eventually reached the state of looseness with which the other bearing started. The loose bearing, on the contrary, practically retained its original clearance. We concluded that the wear of the snug bearing was accelerated because of the absence of an oil-film sufficient for complete lubrication. The loose bearing apparently accommodated an adequate oil-film and the wear was normal. The test was continued for some time and frequent examination showed that the snug or full-fitting bearing continued to wear faster than the loose one. This, we believe, is due to the heavy initial wear that breaks or distorts the ground surface instead of glazing it as seems to be the case when the bearing is assembled with a proper clearance at the start.

Ball-Bearing Clearances

After the transmissions using the roller bearing had been changed to conform to the practice just described, it became apparent that the same gear-noise existed in the transmission used in one of the other Packard models which had the transmission gears mounted on ball bearings. We altered a few experimental ball bearings by deepening the grooves in the races to allow a minute clearance for the balls, instead of assembling them to a good rolling-fit or to the fit of the standard stock. This was done to provide oil clearance, as had been done with the roller bearings. When the loose ball bearings were substituted for the tight ones in noisy transmissions, our previous experience was repeated and the noise decreased. Careful experiments determined the desirable clearance to allow in the races for the reduction of noise, and the manufacturers of ball bearings agreed to supply bearings with various amounts of clearance so that we could determine the requirements of this work.

It was a comparatively easy matter to determine the clearance needed in the case of the roller bearings since diametral clearances could readily be measured. It was found to be difficult, however, to measure radial clearance in the ball-bearing races and we were forced temporarily to determine the degree of clearance by the end-play or axial looseness. A maker of ball bearings pre-



INSTRUCTIONS FOR USE OF CHART

(1.) WHEN THE GROOVE RADIUS IS 52 PER CENT OF BALL DIAMETER

Mark on scale to the left point corresponding to ball diameter of bearing under consideration Mark on scale to the right the radial looseness, place a ruler to cover these points and read off on central scale the axial looseness

(2.) WHEN THE GROOVE RADIUS IS DIFFERENT FROM THE STANDARD 52 PER CENT

Proceed in the same way and multiply the result by the corresponding factor from the following table

Groove Radius, Per Cent	Factor	Groove Radius, Per Cent	Factor
51	0.71	56	1.73
52	1.00	58	2.00
53	1.23	60	2.24
54	1.41		

(3.) The formula from which this chart has been derived is based on the assumption that the axial looseness is equal to the greatest axial displacement of the two races that can be obtained without deforming either the balls or the races

(4.) As all practical methods for measuring axial freedom involve a certain deformation, the measured value is usually greater than the theoretical value here given by about 25 per cent

FIG. 8—NOMOGRAPHIC CHART DESIGNED TO CONVERT THE DESIRED RADIAL CLEARANCE OF A BALL BEARING INTO THE EQUIVALENT AXIAL PLAY

pared the nomographic chart, reproduced in Fig. 8, for the purpose of converting a desired radial clearance into the equivalent axial play. This enabled us to order bearings with a selected radial clearance, which could be held uniform without demanding a finer tolerance from the bearing maker, and cured the gear noise most effectively.

We selected three groups of 10 bearings each and assembled the transmissions with the respective groups. Group No. 1 represented the average clearance in the bearings found in stock; No. 2 contained more radial clearance; and No. 3 still more or about 0.0006-in. radial clearance added to the old standard. These bearings were tested in finished cars by inspectors who regularly passed on gear noise. The nature of the test, its object and the condition of the cars were unknown to them. Our belief in the clearance theory was again upheld by their reports. With one or two exceptions they classified the transmissions as unsatisfactory, fair or good, good or excellent, in the order of the three groups, a fact which shows that the small clearance found in some stock-bearings caused noise and that the noise was eliminated when a slightly larger clearance was provided and strictly maintained.

The amount of clearance to be allowed in any case cannot be determined by rule. We have found that it varies in transmissions similar in design but differing in detail dimensions. Our experience leads us to believe that each mounting must be studied individually, by varying the looseness of the bearing until the suitable amount is apparent. In the ball bearings referred to, a total radial clearance of from 0.0006 to 0.0009 in. was found the most satisfactory.

Backlash and Noise

From the two cases cited we judged that the same reasoning might apply in determining the proper allowable backlash in gear-teeth. Here, again, there is a possibility of not providing sufficient room for a film of lubricant, of squeezing all the oil out of the meshing space and of causing increased noise. Experiments with gears identical in every respect except in the amount of backlash showed that a very evident drop in the noise of the gears was produced when the backlash was increased to a certain point. We now endeavor to hold the backlash of Packard gears within closer limits and work to keep it near to standards, which must be determined for the different designs. Some assemblies will give good results with much more backlash than will others. The high limit is determined invariably by the existence of some rattle; the low limit is determined by the smallest clearance that can be used with oil as thick as that in frequent use in winter weather, without causing growl or bearing noise.

REAR-AXLE BEVEL-GEARS

It has been our experience that whenever we silence one noise in a car, another becomes evident that never caused complaint before. The eventual attainment of silence is the result of a persistent noise-curing campaign which starts with the noise that is most noticeable and works down the line. This method has resulted in finding that the rear-axle noise still persists in most cars though in a much less disagreeable degree.

The spiral bevel-gear, originally introduced by the Packard Company, was a big step in the direction of reducing rear-axle gear-noise. Until other units of the automobile had been perfected to their present state of quietness, we were satisfied that rear axles were about as quiet as they could be made commercially. We are now endeavoring to perfect the assembling of the gear so that the noise of operation shall be reduced still more.

If the presentation of these notes results in a valuable discussion of the subjects treated, we shall feel repaid for our efforts in getting the material together.

THE DISCUSSION

PRESIDENT B. B. BACHMAN:—I have had a suspicion that has been confirmed again and again, that we all have a fetish for close fitting, that we forget many times that oil and oil-films are actual physical things and require space and that, with bearings that just fit, when there is no clearance space between them even for atmosphere, the possibility of getting lubricants into them is reduced to zero.

H. M. CRANE:—I am very much interested in the introduction of loosely fitted gearbox bearings to produce quiet gears and especially in connection with the use of ball bearings. Has any test been made of ball bearings in which the balls and races are fitted tightly but the outer races are fitted loosely in the housing? I do not know that that would give the same result. It probably would not, but I think that it might give the desired cushioning effect and the same quieting result.

R. F. RUNGE[3]:—Referring to Mr. Crane's question, I do not believe that the same results could be obtained by making a loose fit between the outer ring of the ball bearing and the housing. This form of construction may tend to dampen some of the transmitted noise but would be dangerous, as the tendency would be for the outer ring to creep or spin in the housing. This would cause excessive wear, especially where the housing is made of aluminum.

The chart shown as Fig. 8 in the paper is apparently based upon theoretical calculation and is evidently intended to give only an approximate idea of the comparison between axial or end play and radial looseness. Actual measurement of the end play shows that for a specific amount of radial looseness the end play will be from 25 to 30 per cent greater than the amount indicated in the chart. The determination of the radial looseness by end-play measurements is not at all reliable, as the amount of total movement of one ring as compared with the other depends on the accuracy of the contour of the ball grooves, the radius of the grooves, the amount of pressure applied when the measurement is taken and whether the one ring is moved in a plane parallel to that of the other ring.

Considerable variation can be had in all of these items but it is at the same time possible to hold the radial looseness constant. In all of the variations that are encountered, the one that will have the greatest range will come from varying the pressure applied when the end play is measured. Under a very light load, there is excessive end-play due to the elastic deformation occurring at the point of contact of the balls with the grooves. As a result, we have found that where the amount of end play must be considered, its determination can be made only when a specific end-load in either direction is applied. From this explanation it will be seen that the determination of radial looseness by the amount of end play is not very reliable. We have developed machines that will determine either radial looseness or radial tightness within 0.0001 in.

Another consideration that is not always taken into account is the amount of looseness or tightness remaining in a bearing after it has been mounted. We find, from experiments, that where an inner race has a press fit on the shaft, the ball groove of that race will expand approximately 70 per cent of the amount of the press fit. Therefore, considering that a bearing has a radial looseness of 0.0005 in. and the inner race has a press fit of 0.0010 in., after the bearing is mounted it will be actually tight by 0.0002 in., all of the looseness having been taken out by the press fit and 0.0002 in. of tightness put in. Where it has been predetermined that a certain amount of radial looseness is desirable, it is very important to be sure that the amount of final looseness required has not been interfered with by the mounting of the bearing. As it is not general practice to force the outer ring of the bearing into the housing, it is not so important to consider these same conditions as applying to the outer ring, although in such cases as where the outer ring is forced into the housing, that should be taken into consideration. It should also be kept in mind that where those types of bearing that must be opposed to stabilize the position of the shaft are used, it is practically impossible to maintain radial looseness or to know its magnitude.

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[3] M.S.A.E.—Vice-President, S K F Industries, New York City.

A tightly fitted bearing will make more and a different kind of noise than one that is loosely fitted. The tight bearing will also set up a greater noise when inaccuracies in the bearing exist or when foreign matter, such as metallic chips or abrasive matter, is present. The noise in bearings many times is traceable to the presence of foreign matter, and extreme care is taken in the manufacture of the bearings to eliminate this condition and every precaution should be taken by the user to keep the bearings clean in handling or assembling.

When normal or tightly fitted bearings are used, the gear noise, as well as bearing noises, is very efficiently transmitted to the gearcase or the housing which, in most cases, will magnify the noise. It is a well-known fact that when an oil cushion can be provided, it will dampen out part of the noise and vibration. I believe, however, that in the use of the looser bearing for application to transmissions, as the looseness of the bearing will more freely accommodate misalinement of shafts and their deflection under load, greater flexibility plays a bigger part in noise elimination.

G. E. Goddard:—In his experiments with this radial play, did Mr. Brodie discover any appreciable difference in the oil-pumping characteristics of the ball bearing?

In some of our earlier experiences we noticed that some ball bearings pumped oil in a certain direction. I think the problem was never solved in our own minds. We turned the bearings end-for-end and they pumped in the same direction. We tried bearings of different makes, with different ball separators, and that seemed to make very little difference. Would increasing the rolling clearance affect the oil-pumping of the bearing?

I think it is common practice to use an oil spinner or throw-off ring to prevent the oil going through a bearing, but believe it would be rather interesting to hear from some of the ball-bearing men as to whether they have had any experience with curing oil-pumping in annular ball-bearings.

If there is an existing standard on the radius of the groove in the races, is it 7/10 of the ball diameter or do the different makes of bearing have different radius grooves? If they do, would it not be necessary to have a table for each make of bearing?

R. S. DRUMMOND:—We have had some experience in our factory along the same line that Mr. Brodie mentions, concerning the radial clearance of ball bearings. The work that Mr. Brodie mentions in his paper is confirmed by experience in testing gears in our factory. The tests were suggested by gears that were quiet with plain bearings and noisy with other bearings, and were made using Hess-Bright bearings of various radial clearance. I speak of this with reference to automobile transmissions only, as there appear to be other fields of usefulness for ball and roller bearings that are not as yet affected by close fit of balls.

We have upward of 15 different transmissions, representing a number of cars that are produced by men present in this room. These were furnished us by the car builders for test purposes. In every case the bearings in the transmissions require some alteration in the amount of radial clearance in order to operate quietly. Comment has been made about the amount of radial clearance or oil room needed. There are no statistics that are absolute on that subject today. The amount mentioned in Mr. Brodie's paper, 0.0008 in., is approximately correct under average conditions in average transmissions as tested by us. Where there are no means of checking in a laboratory the amount of clearance in the bearing, we have used a very simple expedient. We hold rigidly

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* M.S.A.E.—Vice-President, Gear Grinding Machine Co., Detroit.

with one hand the inner race and the cage carrying the balls and try to slip the outer race over the balls. A crude but very satisfactory estimate of the amount of radial clearance can be made in this way. If the outer race slips in that way, there is normally enough radial clearance. In four noisy transmission jobs investigated in the factories of the manufacturers, we established to their satisfaction that the radial clearance was lacking to an extent that noise developed in the bearings and that noise was eliminated by increasing radial clearance. Our tests confirm what Mr. Brodie has said.

Our experience indicates the same general result from the standpoint of backlash in gears. In cold weather the teeth operate somewhat like individual dashpots. The oil does not always get away with facility at the bottom of the tooth; it has to be squeezed out under pressure and, if there is not sufficient room for the oil to escape readily, a noise will develop. Backlash in various types of gear and in various transmissions depends to a certain extent upon the construction. There is one transmission that requires approximately 0.004-in. backlash to operate with any reasonable degree of quietness. We believe this is due to poor support of the driving pinion. When this support is improved the gears run quietly with greater backlash. Average transmissions show a requirement somewhere between 0.010 and 0.016-in. backlash.

T. C. DELAVAL-CROW[5]:—Mention is made in the paper of the effect of radial clearance in ball and roller bearings on transmission noise. The author's experience indicated that an increase of radial clearance within certain limits reduced the noise. The reason advanced for this is that sufficient space was left for an oil-film between the balls and the races. The experience of the New Departure engineers corroborates the results obtained by Messrs. Crain and Brodie but we cannot agree with the reasons advanced for the reduction in noise, as the very heavy unit-pressures at the contact points of the balls with their races absolutely preclude the presence of an oil-film in the commonly accepted sense of the term and, therefore, it seems that some other reason should be assigned for the reduction of noise.

A study of the noise problems indicates that there are two main causes, namely, noise due to faulty tooth-contact arising either from an imperfect tooth-form or from a misalinement of the transmission parts in assembly, and the secondary cause that might be traced to the inherent structure of the bearings themselves and their mounting. The trained ear can easily distinguish between the two causes of noise, that caused by the bearing being a distinct whistle and that from the gears a growl. In many cases the modification of the bearing structure and mounting will alleviate the noise due to faulty tooth-contact of form. Our experience indicates that there is no general rule as to the degree of modification of the bearing structure that will lessen noise in all cases. Each particular transmission is a study in itself, the design of the transmission housing having a very great influence on the degree of amplification of noise vibrations.

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[5] M.S.A.E.—Chief engineer, New Departure Mfg. Co., Bristol, Conn.

As a general thing, attempts have been made to use ball bearings with no radial clearance, little regard being paid to the effect of the expansion of the inner race due to excessively heavy press-fits on shafts and to too tight fits of outer races in housings. When assembled under these conditions, the bearings themselves will have a distinctive whistling noise due to the vibration set up through the metallic parts under tension, as in the case of a piano string. Noise such as this can be decreased or practically eliminated entirely through the lessening of the press fit on the shaft and the reduction or elimination of the tightness of the fit of the outer race in the bearing housing. It is essential that the outer race be not a tight fit in its housing, due to the fact that it is practically impossible, as a production proposition, to produce housings accurately round; and the extension of out-of-roundness of the housing is transmitted to the ball race of the bearing on account of the deformation of the outer race.

An example of the influence of the housing out-of-roundness on the race ring may be cited from an experiment conducted with a light aluminum housing of an average thickness of 0.1870 in. about the bearing seat which when accurately measured was found to be 0.0006 in. out-of-round in the bore. Into this was inserted with a very light press-fit, a plain bearing ring of high-carbon chrome steel, having a width of 0.6693 in., a section of 0.2960 sq. in. and a base diameter of 2.4410 in., which was lapped to a perfect ring on both the outer diameter and the bore, and accurately measured before insertion into the housing. Upon being inserted into the housing which, as has been previously said, was approximately 0.0006 in. out-of-round, the bore of the ring was found to be 0.0004 in. out-of-round. Upon the reduction of the fit in the housing from a press to a push fit, the out-of-roundness of the ring was eliminated. The expansion of inner-race rings of bearings due to press fits on shafts is likewise often neglected. Investigations have shown that the increase in the diameter of the inner raceway is dependent on two things: the amount of fit used and the relation of the race thickness to the shaft diameter.

A study of the chart at the left of Fig. 9 will show that a press fit greater than 0.0005 in. is detrimental, because the race expansion increases very rapidly above this point, and is also liable to make trouble in assembly due to scoring of the shaft, causing insecure seating and possible misalignment. It is also seen that for a given increase in the shaft diameter the expansion of the cone raceway is greater in the case of the No. 1310 bearing than in that of the No. 1307.

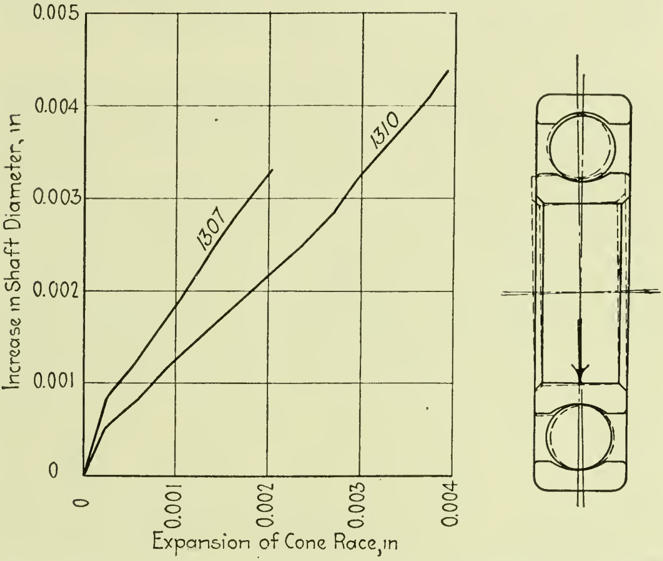

Fig. 9—The Chart at the Left Shows the Relation between the Shaft Diameter and the Expansion of a Cone Raceway in a Ball Bearing, While the Drawing at the Right Illustrates How Much It Is Possible for the Shaft on Which the Inner Raceway Is Mounted To Deflect before the Uppermost Balls Make Forced Contact with the Upper Portion of the Outer Race

Considering the fact that the bearing is used merely to support the rotating members and not as a means of transmitting torque, the use of an excessive press-fit is not necessary. The press fit need be only great enough to assure that the circumference of the shaft is larger than that of the cone bore, in which case no slippage can occur and consequently no wear or peening of the shaft take place.

It is the practice of the New Departure Mfg. Co. in the production of its ball bearings to allow sufficient radial clearance between the balls and their races to compensate for inner-race expansion due to average press-fits not in excess of 0.0005 in. and in addition to allow a very slight excess of clearance to take care of the possible misalignment of bearings in housings or of the transmission shafts in assembly.

A fact that is not as generally recognized as it should be is that in properly constructed ball-bearings provided with slight amounts of radial clearance there is very marked ability to operate satisfactorily under conditions of shaft and housing misalignment. This is due to the fact that in the presence of clearance with other than gravitational load, the inner race and the lower balls will establish perfect contact with the outer race at the lower side, leaving all of the clearance on the upper side between the uppermost balls and the outer race, so that, as illustrated at the right of Fig. 9, the shaft upon which the inner race is mounted can deflect about a suitable angle before the uppermost balls come into force contact with the upper portion of the outer race. Under actual running conditions the clearance between the uppermost balls is greater than under static conditions, due to the deformation of the races and the balls under load, so that the angularity of shaft deflection transmittable under these conditions is in direct proportion to the amount of load on the bearings.

It will be seen that, if the radius of curvature of the outer race approximates the radius of the ball too closely, the ability of the bearing to withstand ordinary deflection misalignments is greatly reduced, in spite of the fact that there may be built into the bearing a relatively large amount of radial clearance.

Our experience has indicated that it is possible to lessen seriously the availability of a given bearing, through the establishment of too close race curvatures, especially in the outer race, to carry ordinary deflections or misalignments without excessive noise. While, in a properly designed and manufactured ball-bearing, a limited amount of radial play is desirable for the reasons given above, it can be seen readily that excessive radial clearance may become detrimental to the load-carrying ability of the bearing in that when there is excessive radial clearance the diameter of the ball race in the outer race is materially greater than the sum of the diameters of the ball race of the inner race plus two ball diameters, causing the radial load of the bearing to be carried on two or three balls rather than on half of the balls in the bearing.

As stated by Messrs. Crain and Brodie, it is practically impossible to measure radial play. We therefore advocate that the measurement of end play under a known load be used as an indication of the radial clearance existing in the bearing. It is well to remember that this means of measuring is only an indication, as the mathematical relation of the radial clearance to the end play is dependent on the percentage of the ball diameter to which the race radius is ground and a very slight change in this percentage makes a relatively large change in the end play obtained for a given amount of radial clearance. The race radius on all high quality ball-bearings is ground by a generating grinder and any change in the race diameter from the standard set-up causes an equal change in the race radius, therefore causing a variation in the end-play relation to the radial clearance.

SOME CAUSES OF GEAR-TOOTH ERRORS AND THEIR DETECTION[1]

By K L Herrmann[2]

The different gear noises are classified under the names of knock, rattle, growl, hum and sing, and these are discussed at some length, examples of defects that cause noise being given and a device for checking tooth spacing being illustrated and described. An instrument for analyzing tooth-forms that produce these different noises is illustrated and described.

Causes of the errors in gears may be in the hardening process, in the cutting machines or in the cutters. A hobbing machine is used as an example and its possibilities for error are commented upon. Tooth-forms are illustrated and treated briefly, and the hardening of gears and the grinding of gear-tooth forms are given similar attention.

Motor-car production-men, as a rule, do not lay claim to being specialists in all the various arts and sciences that enter into the finished motor-car; so, in the matter of gears, we do not pretend to know all the details that enter into their design and production. Very few of us are familiar with the mathematics relative to the involute gear-tooth forms that the engineering fraternity stresses considerably in connection with gear noise. We are much more familiar with the noises of the finished product, which are recognized as knock, rattle, growl, hum and sing. When these occur, naturally those men who have made a life study of gear subjects are called in, and they make recommendations.

As a rule our gear experts offer widely varying remedies for the same cause. If one is using a 20-deg. pressure-angle, we can advise a 14½-deg. angle and have a large number of supporters in both the engineering and the production fields. If both 20 and 14½-deg. angles have been tried, we can easily advise a stub tooth or full length. If all six combinations have been tried, it is easy to recommend topping-off or no topping-off; and, if this does not work, we recommend a different *steel*, *cutting machine* or *cutter*. While all of this goes on, sufficient time has elapsed to start all over, with the first recommendation. Of all of these recommendations we find many past and present supporters of national reputation.

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[1] Detroit Production Meeting paper.

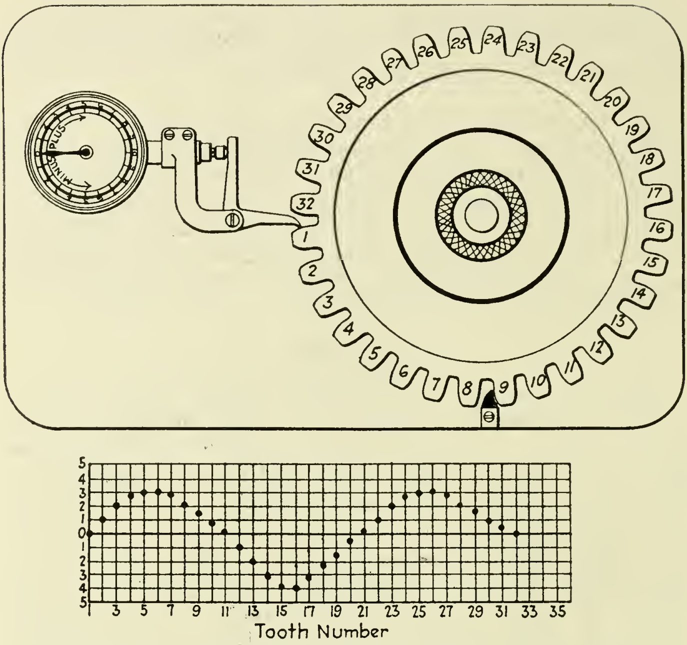
[2] M.S.A.E.—Engineer, Studebaker Corporation, Detroit.

It is the purpose of this paper to show that production variables have a much greater influence on gear sounds than changing pressure-angles, steel or tooth-form details; also, by showing the errors present, to obtain definite help from the gear-cutting tool and the machine designer. We will confine this discussion, for the time being, to the transmission, which is the simplest type of gearing used in the motor car.

GEAR NOISES

We have already referred to the various kinds of gear noise. The first is a knock that might be caused by a nicked tooth or a single tooth of a pair of running gears that are in mesh. If this single tooth or nick happens to be in the transmission constant-mesh pinion and it is driven at 1000 r.p.m., there will be 16 distinct blows per sec. Elementary physics shows that, up to this point, these blows can be distinguished by the human ear as individual blows. With this same gear, rotating at 1000 r.p.m., if there are two nicks and they are a uniform distance apart in the gear, the gears will have 32 knocks per sec. This noise still will be distinguishable as individual knocks by some ears, but by others it will be noted as a distinct tune. However, if the two nicks happen to occur in this gear an uneven distance apart and the transmission is speeded-up, one can note readily the place where the gear noise changes from an individual knock into a sound that is usually designated as a rattle.

It is evident that influences similar to the nicks referred to above can be produced by inaccurate conditions in the gears. For example, should the tooth spacing in the transmission drive-pinion mentioned be such that the driven gear, instead of rotating at a uniform speed, is forced to increase and decrease its speed at every revolution of the pinion, very similar sounds will occur. If on one side of the driving pinion the teeth are 0.015 in. ahead of their proper position on the periphery of the pinion, the driven gear must gain and lose a corresponding amount in its steady motion. In so doing, it may strike on the back of its tooth rather than on the driven side, especially when idling, and produce a series of blows that further increase and confuse the rattle.


PLUS
MINUS
1
2
3
4
5
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7
8
9
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17
18
19
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21
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27
28
29
30
31
32
5
4
3
2
1
0
1
2
3
4
5
1 3 5 7 9 11 13 15 17 19 21 23 25 27 29 31 33 35
Tooth Number

Fig. 1—Device Used for Checking Tooth Spacing and Underneath a Chart Showing the Results Obtained

It is not sufficient to check gears for spacing error from tooth to tooth. It is very desirable to check the accumulated error of a number of teeth, because a gear may vary 0.001 in. from tooth to tooth. With eight successive teeth each gaining 0.001 in. on the side of the gear and a similar number of teeth that may be losing 0.001 in., a total error of 0.016 in. might be imparted to the driven gears.

Fig. 1 illustrates a very simple device that has been used for checking tooth spacing. The gears are mounted on a bushing and one tooth comes against a stop. A dial indicator is arranged so as to be in contact with some tooth one-fourth, one-third or one-half way around the gear. When the dial indicator is set at zero, with the tooth against the stop at any one point, the distances between the two points can be measured and, if the gear be correct for indexing, placing any two of the teeth in the gear in similar positions should not cause the dial indicator to vary, especially if the gear runs true.

When the gear is first put on the indicating apparatus, the dial indicator is set at zero. We then put a mark at zero on the chart in Fig. 1 for tooth No. 1. The next step is to index the gear around one tooth. Any reading obtained is marked above the tooth number in the vertical line. We next index the gear around to tooth No. 3 and again mark the dial-indicator reading opposite the number of thousandths of an inch that it may show. The gear is then indexed to teeth Nos. 5, 6, etc., until all the teeth on the gear have been indexed.

For the purpose of record, we now have a chart showing the accumulated variables. It will be seen from Fig. 1 that at no point is the spacing variable as great as 0.001 in. between any two teeth, but it can be in error a total of 0.008 in. or more when the error between the several teeth has accumulated. A better visual demonstration of this condition occurring in gears is made by

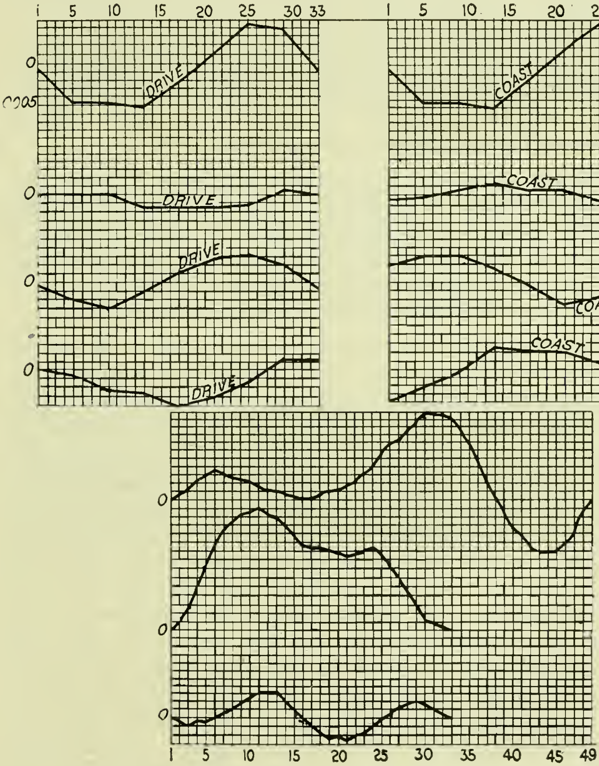

Fig. 2—Chart of the Variations in a Finished Gear

means of a gear-tooth-form projector. When a gear-tooth form is projected upon a screen by this device, it will be noted that the magnification of the shadow is 100 to 1 and that, for every inch on the screen there is at least a 0.001-in. error somewhere in the gear. The shadow on the screen also shows the variation in uniform movement of the driven gear due to an error of this kind; that is, the driven gear, instead of having its tooth in the position of the outline on the screen, has been forced to advance a number of thousandths of an inch. It will be noticed further that this advance and retardation does not occur uniformly; that is, the advance may be confined to a very small number of teeth, remain there for a certain length of time and then be retarded slowly. A gear in this condition will give a rattle very similar to that which might be produced by unequal spacing. This condition can be studied best by charting it as described.

A number of variations as they occur in the finished gear are shown. Fig. 2 shows, first, a rapid drop, then a flat portion showing no change, then a rapid rise, another flat and then a return to the starting point. These curves require very little explanation.

It requires considerable imagination to determine just what happens when a driven gear such as those described is meshed with a driving gear having teeth misplaced as shown in Fig. 3; yet this occurs in daily practice. Fig. 4 shows a number of gears and pinions none of which has an index error of more than 0.005 in.

In addition to the two gears having a rather irregular action between themselves, their increasing and decreasing movement is carried on to the countershaft and the idler, which often have similar defects in themselves. Accumulating errors in gears often cause the fourth gear in a train to be as much as 0.025 in. away from its correct position and this change occurs in varying amounts depending on the number of teeth in the gears of the gear train. From this it is seen readily that the countershaft does not rotate smoothly and that the gear on the other end of the countershaft, in addition to the variable movement given it by the errors in the first two gears, imparts its error to the gear meshing with it. The action of the fourth gear will also lack uniformity in addition to that imparted to it by the driving gear, to the extent of the error on the fourth gear. If the ratios between the gears referred to are in direct proportion,

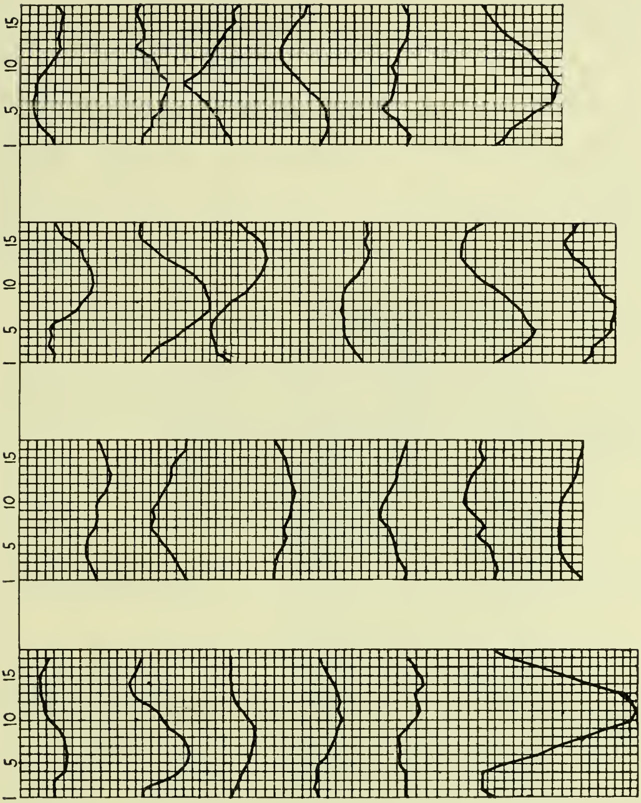

Fig. 3—Errors in the Spacing of the Teeth of a Driving Gear

such as 2 to 1, these several shocks will occur uniformly, the error of the first uniformly with each revolution of the respective gears; however, if the ratios should be odd, such as 16 to 31, it will be difficult to estimate the change of speed that will take place.

Hum or sing is not nearly so difficult to analyze as the matter of rattle in a transmission gear. With this same transmission run at a speed of 1000 r.p.m. at the pinion shaft, if the pinion should happen to have 16 teeth, it will be found that 250 teeth per sec. go into and out of mesh. If the teeth are not correct in shape and the gears are under a slight load, there may be 250 blows per sec. under certain conditions, which we are told corresponds to the tone of middle C on the piano. Should fewer teeth go into and out of mesh, and this may be caused by a slower speed, a much lower pitch can be produced. In a similar way, because of the speed reduction

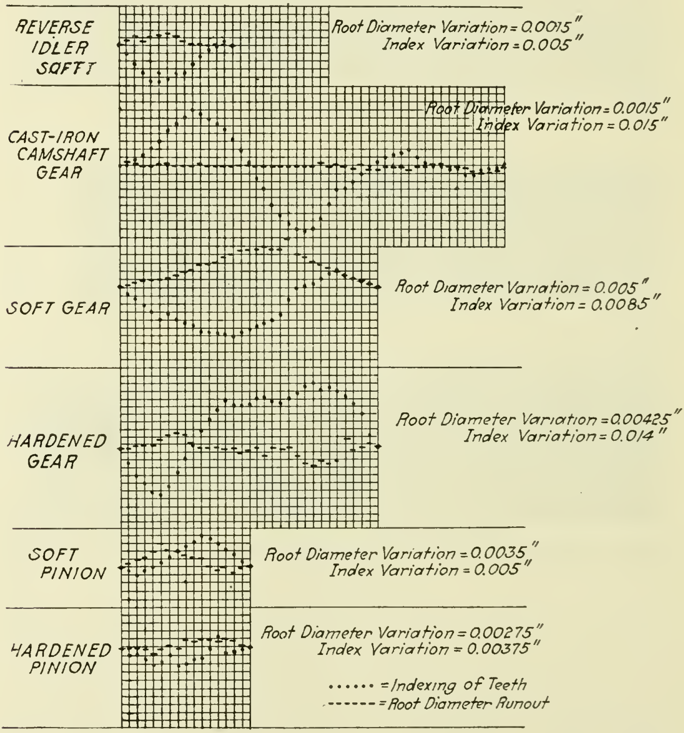

Fig. 4—Chart Obtained from a Number of Gears and Pinions None of Which Has an Index Error of More Than 0.005 In.

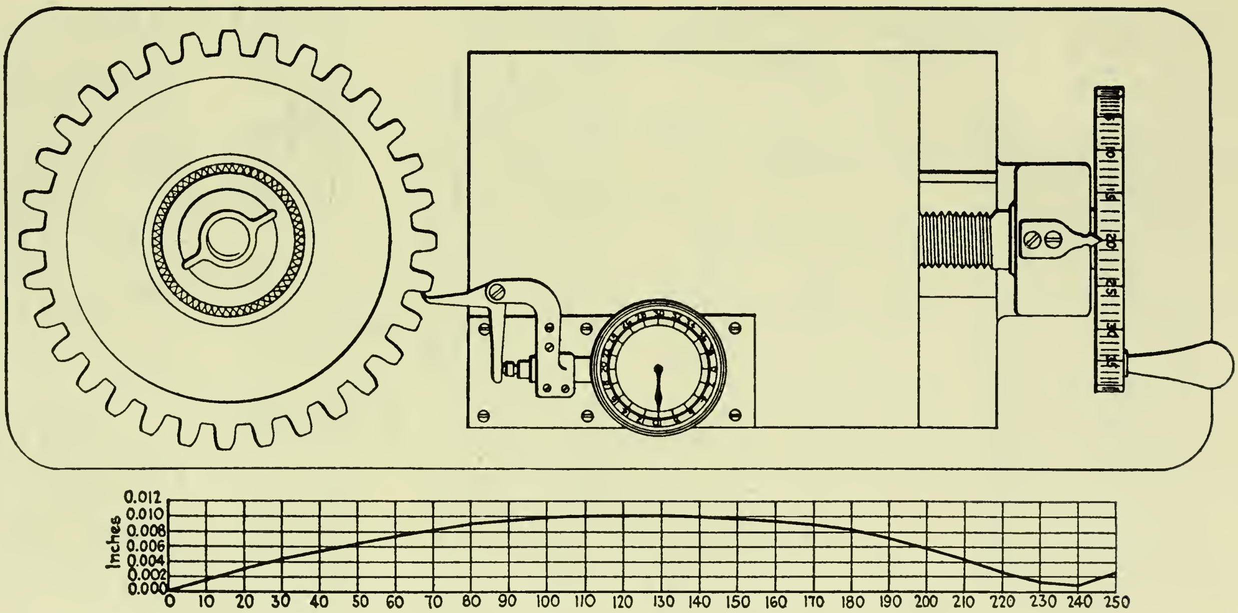
5
10
15
20
25
30
35
Inches
0.012
0.010
0.008
0.006
0.004
0.002
0.000
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
170
180
190
200
210
220
230
240
250

Fig. 5—Instrument for Obtaining an Analysis of Tooth-Forms Producing Noise and a Chart Showing the Outline of a Particular Tooth

that occurs in the usual type of transmission between the drive pinion and its countershaft, the tone produced by the reverse idler is very low and, instead of producing a hum or sing, it will produce what we usually call a growl. Errors in the sliding gears, because of their higher speeds, will produce higher pitch growls and approach a hum.

A great many instruments have been developed for the purpose of analyzing tooth-forms producing these sounds. The one that we have worked out and have chosen to use is shown in Fig. 5. It consists of a dial indicator mounted on a guided slide. We place the gear in a definite position with respect to the indicator, start at the point of the tooth and set the indicator at zero. The slide is then moved toward the gear 0.010 in. and the indicator-reading marked on the chart shown in the lower portion of Fig. 5. The slide is then moved 0.010 in. more, the reading is marked again, and this is continued until the bottom of the tooth is reached. By taking the gear that has just been charted off the bushing and placing another gear in its place, other tooth-forms will be compared with the first.

The charts reproduced in Fig. 6 show some variations occurring in production, these particular charts having been made from gears with which special care had been taken to secure good gears. This was done by men who were not previously familiar with the method that was going to be used for the gear inspection. Fig. 7 shows some curves that occur in production from the same hob, and also curves made at different hours of the day.

CUTTING MACHINES

The causes of the errors referred to are various. Some of them occur in hardening, some in the cutting machines and some in the cutters. We have found these errors in all of the types of machine that we have used. For the purpose of this discussion we are selecting a hobbing machine.

The hob is a generating tool that produces a gear such as is shown in Fig. 8. It will be seen readily that if all tooth-heights of the hob are the same, each hob-tooth generates, roughly, a flat in the tooth-form. If any of the teeth in the hob is high, a wider flat will be produced as shown in Fig. 9. Should the tooth-heights be correct and the hob be mounted in the machine with a run-out, a leaning tooth can be produced, depending on the sidewise

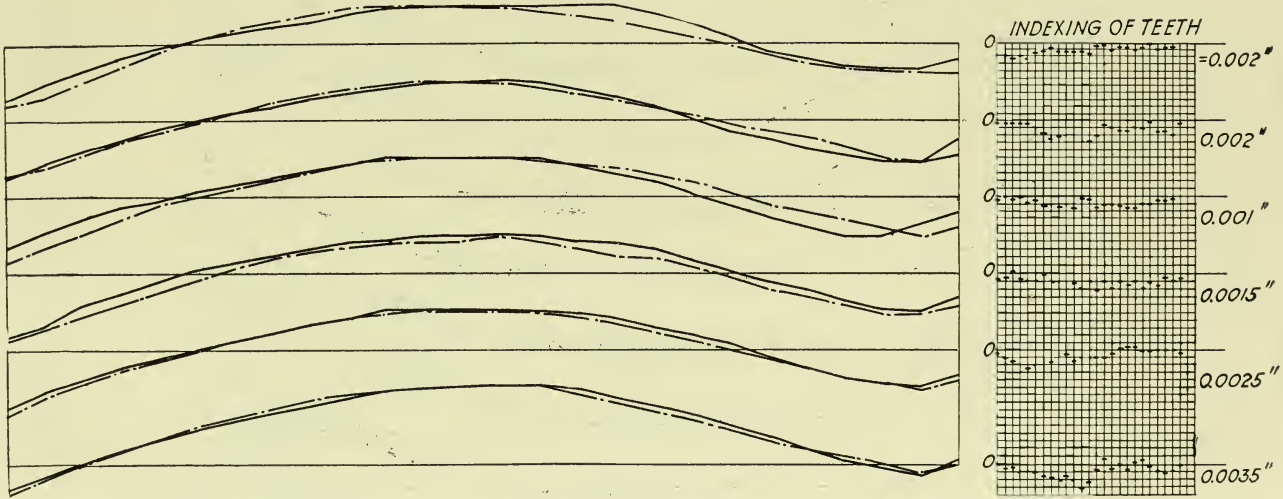

Fig. 6—Charts Showing Some of the Variations That Occur in the Production of Gears

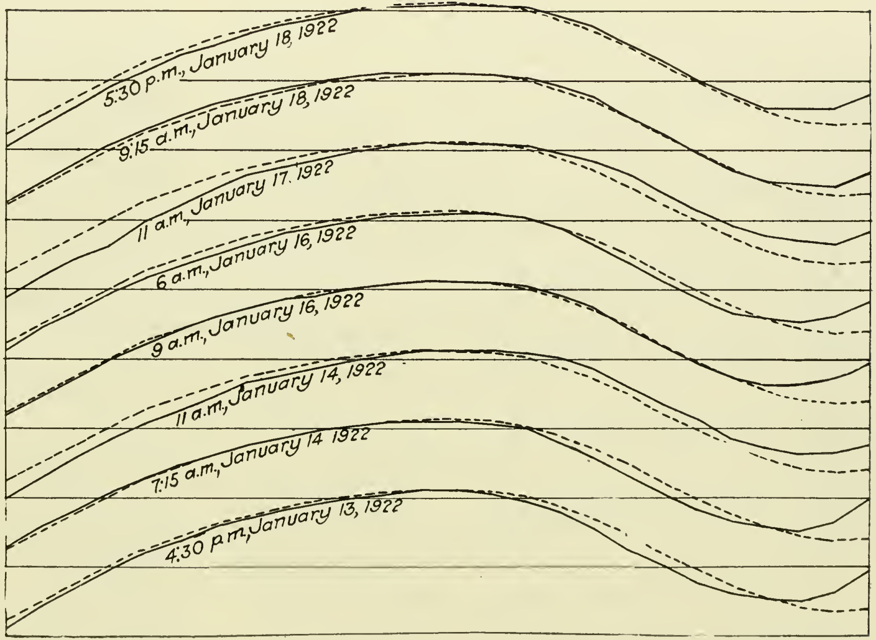

FIG. 7—VARIATIONS OCCUR IN THE PRODUCTION FROM THE SAME HOB ON DIFFERENT DAYS AND ALSO AT DIFFERENT HOURS OF ANY ONE DAY

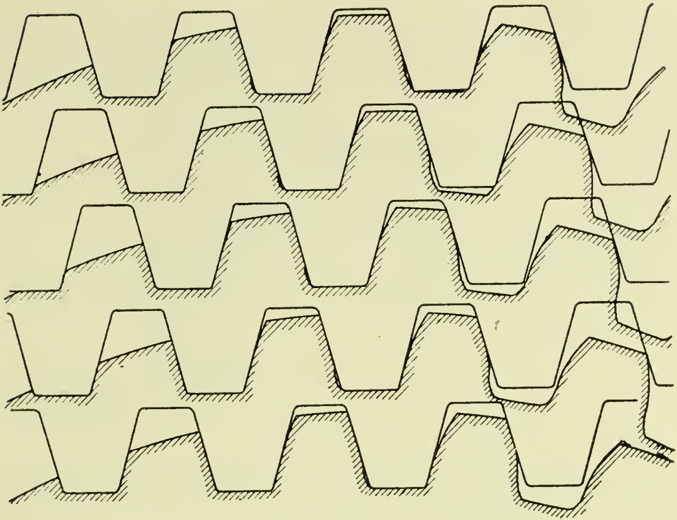

Fig. 8—Outline of a Gear Produced by the Hobbing Method

setting of the hob with the gear. Also, should the hob be correct and the end-thrust collar in the hob spindle be out of parallel, giving the hob a slightly reciprocating motion with each revolution, an error can be produced that may compensate for the hob run-out or may add to it. Should the thrust collar at the rear end of the spindle

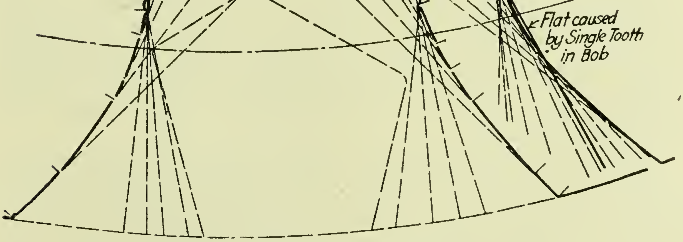

Fig. 9—A Single High Tooth in a Hob Will Produce a Wider Flat in the Gear Than if All the Teeth Are of Uniform Height

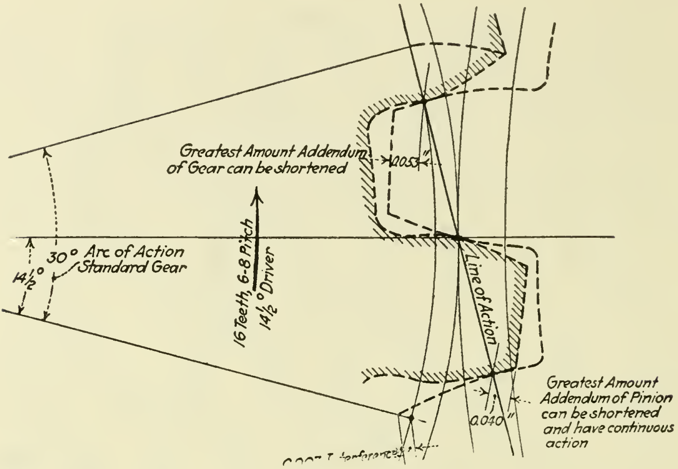
Greatest Amount Addendum of Gear can be shortened
0.053"
30° Arc of Action Standard Gear
14½°
16 Teeth, 6-8 Pitch 14½° Driver
Line of Action
0.040"
Greatest Amount Addendum of Pinion can be shortened and have continuous action

FIG. 10—PRELIMINARY LAYOUT THAT IS MADE TO DETERMINE THE LENGTH OF TOOTH NECESSARY TO GIVE A COMPLETE ARC OF CONTACT

be adjusted loosely so that the spindle may have end-play, the hob, as it cuts on one side, will be forced over and then back with each tooth of the gear and produce corresponding errors in the tooth-form. Again, if the gears in the hob-grinding spindle have inherent index-errors, or should the gears driving this gear be concentric or improperly spaced, their errors will be transferred to the different teeth of the gear being cut. Another important element in connection with the hob is the fit of the hob spindle. We will all agree that, should the hob spindle be tight for a certain portion of the revolution and loose for a certain other portion of the revolution, a sagging will occur in the driving-gear train which will be very detrimental to the tooth-form. Some hobbing machines are built so that the bevel gears in the hob drive-spindle give a thrust in the opposite direction to that given by a spiral pinion driving a hob-spindle gear. This permits a back-and-forth movement of a hob-spindle drive-shaft and sometimes leaves its impression on the tooth-form.

Without going into the details concerning the other gears in the hob-spindle train, we might consider the influence of the thrust collar and the fits of the work spindle. In most hobbing-machines the bearings are kept fairly tight, and a great many operators insist that the hob spindle be kept warm. This also applies to the worm-shaft driving the wormwheel and, to a certain extent, to the work spindle. Unless the machines are extremely well adjusted, the thrust collars on the spindles so fitted score very easily and cause the spindles to be tight or loose, depending on various portions of its revolution. This causes a corresponding sag in the gears driving the index wormwheel and seems to be the main cause of the index errors already referred to. Another source of error is looseness in the gibs of the hob saddle. It is very difficult to move the hob slide across the face of the tooth without having some play between the gibs. This amounts, in a very similar manner, to the

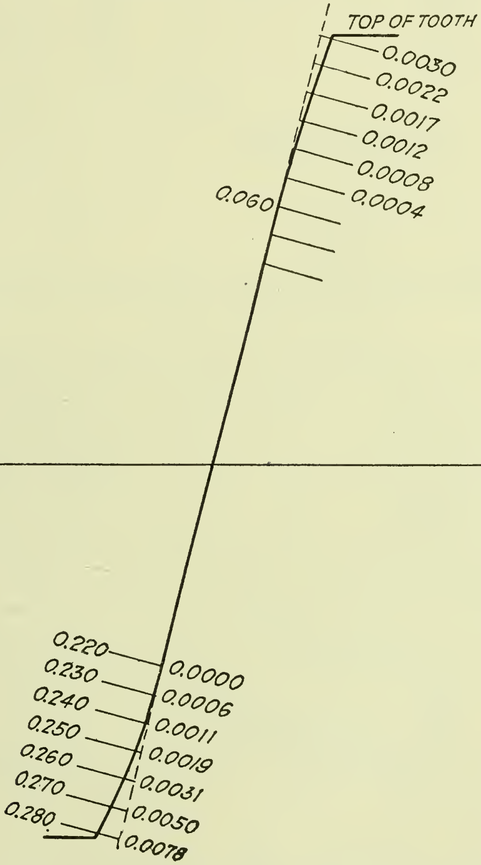
TOP OF TOOTH
0.0030
0.0022
0.0017
0.0012
0.0008
0.0004
0.060
0.220
0.230
0.240
0.250
0.260
0.270
0.280
0.0000
0.0006
0.0011
0.0019
0.0031
0.0050
0.0078

FIG. 11—OUTLINE OF THE HOB FORM SELECTED FROM THE LAYOUT SHOWN IN FIG. 10

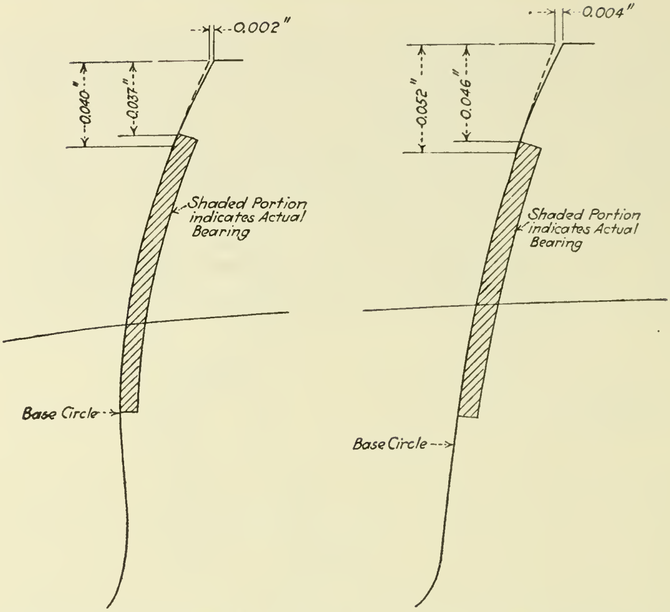
0.002"
0.040"
0.037"
Shaded Portion indicates Actual Bearing
Base Circle
0.004"
0.052"
0.046"
Shaded Portion indicates Actual Bearing
Base Circle

Fig. 12—Tooth-Forms of a Pair of Gears That Are Intended To Be in Contact with Each Other, as Obtained from the Hob Outline Shown in Fig. 11

error that is obtained by having end-play in the hob spindle. Another contributing error of tooth-form is depth of cut. There will be considerable error in the tooth-form if the hob is sunk several thousandths of an inch too deep. Should the hob be straight-sided, the pressure-angle will increase with the depth of the cut and decrease as it is raised above the pitch-diameter. Another factor having considerable influence is the outside support for the hob spindle. We have had considerable difficulty in placing this outboard support of the hob spindle back in exactly the same place, giving us exactly the same condition of hob spindle as before.

When the work spindle is caused to rotate ahead of or behind its proper position, we necessarily have certain tooth-form errors in addition to index errors, and also in addition to those produced by the hob, its spindle and driving mechanism. There are conditions under which some of the errors referred to are counterbalanced by other errors. However, there are also conditions in which these errors accumulate. Considering the number of gears in a hobbing machine and the number of possibilities for errors outside of these gears, it is largely a matter of chance whether suitable combinations can be obtained to produce proper tooth-forms.

TOOTH-FORMS

Having all these liabilities to error in mind, the question often arises as to which is the noisiest tooth-form. Of the various kinds of gear that we have been able to cut and that we have had cut for us by a large number of different manufacturers, we have found that the errors from a general definite shape have apparently more to

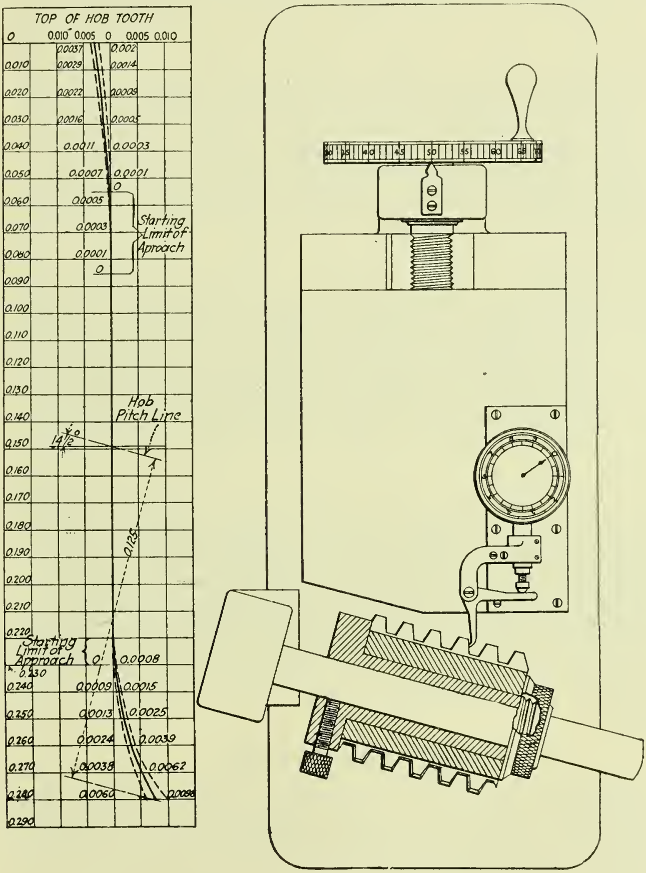
TOP OF HOB TOOTH
0
0.010
0.005
0
0.005
0.010
0.0037
0.002
0.010
0.0029
0.0014
0.020
0.0022
0.0009
0.030
0.0016
0.0005
0.040
0.0011
0.0003
0.050
0.0007
0.0001
0
0.060
0.0005
0.070
0.0003
Starting Limit of Aproach
0.080
0.0001
0
0.090
0.100
0.110
0.120
0.130
0.140
Hob Pitch Line
0.150
14½°
0.160
0.170
0.180
0.125
0.190
0.200
0.210
0.220
Starting Limit of Approach
0
0.0008
0.230
0.240
0.0009
0.0015
0.250
0.0013
0.0025
0.260
0.0024
0.0039
0.270
0.0038
0.0062
0.280
0.0060
0.0096
0.290
35
40
45
50
55
60
65

Fig. 13—Typical Diagram That Is Furnished Suppliers of Hobs Giving the Outline of the Hob Form and Apparatus Used for Checking Hobs after Their Receipt

do with noise than any existing type of gear. The nearer to being correct a 14½-deg. pressure-angle-tooth gear is, the quieter it will be. We have never yet been able to secure 14½ and 20-deg. pressure-angle gears that had similar errors. The indications are, however, that the differences in sound due to the differences in the pressure-angle are very slight when compared with the differences in sound due to different errors in the same gear. We are using 14½ and 20-deg. pressure-angle gears regularly in production, and there seems to be little to choose between them.

The details of tooth-form are of some importance. Continuing with transmission gears, Fig. 10 shows the first layout that we make. This is with a view to determining the amount of tooth length necessary to give a 100-per cent arc of contact. With this information in hand, we select a hob form such as is shown in Fig. 11 and, using this on paper, we roll out two tooth-forms as shown in Fig. 12, one for each of the gears that are intended to be in contact with each other. The next step is to roll these gears on each other to determine the interferences, if any, and the amount that they are topped-off; then, if necessary, the hob form is modified and the same procedure carried through. When the hob form is established on paper in this manner, it is charted as shown in Fig. 13 and the hob supplier is asked to conform to this shape. Definite tolerances are given for the amount of variation from this form. On receipt of the hob, we inspect this form on a hob-checking apparatus very similar to that which we use for checking gear-tooth forms, a drawing of which is shown in Fig. 13. If these conform to our standard requirement, it is expected that the hob will be satisfactory. We do not require any test of the hob in a cutting machine, because of the large number of errors that will be introduced by the machine, either in correcting hob errors or resulting in having an apparently correct hob rejected without cause.

HARDENING OF GEARS

Relative to the errors produced by hardening, we have prepared a number of charts showing the condition of the gear in the green and the condition of the gear in the hard. Fig. 14 shows 24 such gears and the variations occurring in them. Similar variations occur in gears mating with these gears. It will be seen that the

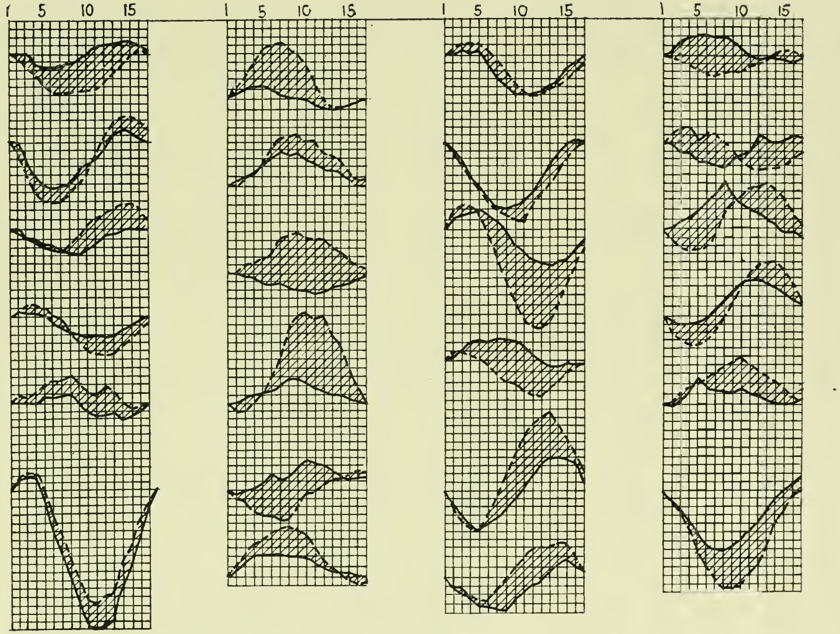

Fig. 14—Chart Showing the Variation in 24 Gears, Before and After Hardening

cutting errors are considerable and that, at times, the hardening errors compensate for the cutting errors, while at other times the hardening errors accumulate in the same direction.

Prof. John J. Keller gave a very interesting paper before the American Society for Steel Treating recently on why steel warps, showing what happens in a piece of steel when it is cooled and quenched. My impression from this paper is that freedom from warpage in steel is a matter of uniform hardening and cooling; also of removing all forging strains before machining. The question whether oil-treated steel is better than carburized steel is still unanswered as regards warpage. We have hardened more than 5000 gears of different brands of steel and carefully checked them. We find that there is very little difference in the warpage under the same hardening conditions.

Grinding Gear-Tooth Forms

The necessity for grinding gear-tooth forms depends largely on the ability to cut and harden gears, maintaining definite shapes. However, there is a large difference in the number of rejections that we have from gears ground by different processes. Our reports at this time show that out of 5000 gears ground by one method we have had a 14-per cent rejection. This is slightly greater than that which we have had from the hob gear without grinding. By another method of grinding of a similar number of gears, we have had less than a 0.5-per cent rejection, as well as more satisfactory gears. In the first case, four gears of the transmission were ground, and in the second case only two gears of the transmission were ground. All transmissions were passed by the same inspector and inspected to the same standards. We, of course, are looking forward with great interest to the continuation of our experiment on gear-grinding.

THE DISCUSSION

J. F. Rand:—Were the gears that we have been discussing finished in one cut? If not, how much material was left on them for the finisher? What type of machine finished them, a hobbing or a gear-shaping machine? We find it advisable to attempt to build a special machine of our own to overcome the indexing trouble mentioned.

O. Lindberg:—We have found it impossible to produce an accurate gear with a gear-hobbing machine. We have used gear-hobbing machines only for roughing purposes, and use a gear-shaping machine for finishing.

C. F. SCOTT:—Many of the British and French cars are built with relatively small engines for economical reasons. The cars are fitted with four-speed gearboxes and much gearshifting is necessary, which might not be acceptable in this Country. These gearsets are, some of them, direct-drive on third, and some drive direct on fourth speed but, in any event, at one of the two top speeds they would be comparatively noisy. Would the noise at either top speed be considered excessive in accordance with American standards, or is the transmission fairly quiet? If so, how do they make it quiet?

MR. RAND:—What is Mr. Herrmann's opinion regarding just how far one can go with spiral angles? In other words, supposing a 30-deg. spiral angle on a given diameter of gear of a certain pitch, is it an advantage to go over 30 or 35 deg.? What is the limiting factor on that spiral angle?

A. J. BAKER[3]:—The essential point Mr. Herrmann brought out is how difficult it is to maintain the proper operation of the gear-cutting machine. When it is considered that this machine, no matter of what type, is composed of two rotating members, each subject to eccentricities due to the initial errors brought about during their manufacture, and that those errors can be added to by temperature conditions, stresses, material variations and relative dullness of the cutting portions, we are forced to the conclusion that the whole principle is wrong.

It is of little use to make a statement of that kind unless some constructive criticism is offered. Do the gear-cutting-machine builders give any consideration to the use of a single-point finishing-tool, such as a broach, in conjunction with the rotating work, so as to eliminate all of those points that give trouble? I have in mind that a single-point tooth for the actual finishing tool can be made far more accurately than it is possible to produce a number of cutting tools, each of which has a different relation to the true axis of rotation. That is applicable, I believe, to only a straight spur-gear, but we should all be happy if we could eliminate some of our spur-gear troubles.

MR. DRUMMOND:—The indicating apparatus shown by Mr. Herrmann is well known to me. I admire the work that he has done. He is more or less of a pioneer in the study of tooth-form. The efforts to develop mechanisms for measuring gear teeth are worthwhile. Many types of measuring apparatus are being offered today. In the interest of good gearing, they should be given careful attention because they are worthwhile.

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[3] Research engineer, Willys-Overland Co., Toledo.

I am also interested in grinding operations and in tooth-form error or distortion. The transmission Mr. Herrmann is working on is one of the most difficult transmissions in America to make quiet. It is a combination of 16 and 32-tooth gears and a rear gear having 13 teeth that drives a 14-tooth gear. The teeth are stub teeth and the pressure-angle is 14½ deg. If you will figure it out and then cut some of these gears in your own factories, I think you will look at the charts again with more interest and pay greater attention to the subject of measuring gear-tooth form. The transmission in question is hard to build, but this is being done successfully.

Mention was made of a machine that would have a broaching action. Gear teeth can be and have been broached. We are using the shaping-type machine with a grinding-wheel form to fit the two sides, which is more-or-less similar to the broach in its action. Regardless of our own individual method of doing this work, we think that the future of the industry lies in the direction of doing final work after the distortion takes place. We are not opposed in any way to doing it by any other method, but I think there will be difficulty in making hardened gears until we do final work after distortion.

The variation in gears that Mr. Herrmann speaks of sometimes takes peculiar form. We have on record a gear that had a number of teeth, going half-way round the gear, with an index error as high as 0.015 in., but gear noise in the transmission was remarkably slight.

K. L. HERRMANN:—It is true that grinding can be resorted to. We have had rather wonderful success with such a machine for the last 2 months. In this particular grinding process, however, it is necessary that the gear be fairly accurate; otherwise the output of the machine is very limited. We are getting gears with less than 0.4-per cent rejection when ground on this machine. However, grinding alone does not solve the problem. We do not need to harden some gears. Some gears are spiral. They involve a somewhat more expensive process than we can afford to use in connection with a medium-price car. Spiral-bevel-gear grinding is unknown to us now.

The gears mentioned in the paper were roughed-out first and then finished in a more accurate machine that is more carefully maintained. It is possible to hob an accurate gear. Gears can be hobbed as accurately as they can be cut on a shaping machine.

The practice in Europe is entirely different from that in America. Some years ago I worked as a toolmaker at the Napier plant in London. When producing only a few cars per week, many things can be done that cannot be done in large production. European gears are much quieter. The work put upon them makes them so.

I do not know whether a 30-deg. spiral angle is better than 45 deg., which we use to a large extent. The spiral angle, as well as the pressure-angle, and tooth heights of full or short length seem to have very little to do with the matter if the other errors are within proper limits. I prefer the steeper angle.

As to the necessity for a new gear-cutting machine, most of the important gear-cutting-machine builders are designing new machinery.

Silent chains help to reduce noise in a motor car. They offer many problems. I refer you to the service divisions of the companies that use them. Accuracy of gear-tooth form is necessary; otherwise silent chains are noisy. The same principles that have been discussed here in connection with gears can be discussed in connection with silent chains.

After we do the best we can with gears and get them as nearly right as we can, the gear situation costs the motor-car companies $11,000 per day. We estimate that this figure represents the lowest possible amount that is being spent on this problem.

A. B. REYNDERS:—One of the principal reasons for using plain bearings on lighting generators is the quieter operation in comparison with that of ball bearings running at high speeds.

Owing to the fact that no absolute standard of noise is available, there are about as many different opinions with regard to quiet operation as there are persons judging the apparatus. To eliminate this difference of opinion we should have some sort of noise-measuring device or some kind of standard. At present we select a machine which, in the opinion of a number of people, is considered quiet. This machine is set aside as a standard and production is obtained by making comparisons with this standard. Is a more accurate method available at the present time in the manufacture of automotive parts?

E. PLANCHE:—It is a recognized fact that gear noise is very intimately connected with the pitch sizes of gears. A noise is created by a metallic vibration. This metallic vibration is produced by the impact of one gear tooth against another. The greater the pitch for a given diameter, the greater is the interval between each impact for the same speed. I am sure that, if the engineers would turn their thoughts to the utilization of a smaller tooth-pitch, widening the gear if necessary to preserve the strength, many of our gear problems would be eliminated or greatly decreased. To confirm this opinion, we have only to study the steam turbine in which the use of small-pitch spiral-type gears of great width has been very successful in transmitting power with the minimum amount of noise.

SELECTION OF MACHINE-TOOLS[1]

By A J Baker[2]

The problem of determining when to make a change of equipment by substituting new machine-tools for old, or special machines for standard, is carefully investigated. The fact that most manufacturers already have a surplus of machine-tools on hand on account of the demand for excessive production caused by the war makes the problem one not of providing for increased production but of decreasing its cost. The advantages and disadvantages of both special and standard machine-tools are weighed and the conclusion is reached that, although the ability of a special machine to produce pieces in fewer seconds is usually greeted with enthusiasm, other considerations such as the possible changes of the design of the pieces to be made, the inability to secure repair parts quickly, the dearth of skilled labor and the waste caused by employing inefficient help may make the change inadvisable. A method of analysis is given, by which an executive can determine how many cars of a particular model must be produced before a change of equipment can be justified.

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[1] Detroit Production Meeting paper.

[2] Research engineer, Willys-Overland Co., Toledo.

I propose to lay down some general principles by which equipment can be scrutinized and the desirability of installing it determined. The title of this paper indicates machine-tools only, and since the application of these principles will be found to be greater with machine-tools than with any other type of equipment, we may let the title stand. I shall make a difference, however, because an executive, when equipping a plant, must select some items of equipment, not necessarily because he can effect a saving by them, but because he cannot produce a commercial success without them.

It is just as important for an automobile to have a body as to have a differential gear, but since the differential gear can be produced in a variety of ways and the sheet metal of the body in practically only one, the question of proper selection becomes much more important on the smaller and less expensive equipment required for the differential than for the heavy and expensive presses required for the body. Generally speaking, these principles apply to the selection of such machinery as lathes and vertical drilling, grinding, broaching, shaping, gear-cutting and milling machines, standard lines of wood-working machinery, hammers and the smaller sheet-metal presses. And in outlining those machines we must consider also those that were specially developed. Although they are described under other and special trade names, yet in view of the work produced these machines still come under the same general classification that is applied to the simpler standard machines.

A primary consideration that an executive must give to any purchase, be it design, material or equipment, must of course be its suitability for the purpose intended; another is the availability of a source of supply. Touching for a moment on this second point, we may look into the source of supply of the machine-tool industry during the last 10 years.

One extremely favorable aspect of the matter is that there is no apparent tendency of the machine-tool industry to become monopolistic in character. It is true that an association exists and it is also generally true that such associations ultimately must be paid for by the consumer. Many examples of this sort no doubt present themselves to you. However, since the manufacture of machine-tools apparently has always attracted a number of new devotees each year and since the various establishments range in size from those employing 50 men to those employing from 3000 to 4000, we can feel reasonably well assured of a diversity of interest and of sufficient competition to make it appear unlikely that any association can dictate to us as to the equipment we shall buy or the prices we must pay. In addition to this, the very remarkable growth of the machine-tool industry must not be forgotten. This Country is, without doubt, a greater producer of small and medium-size machine-tools than is any other. It does not stand proportionately so high in production of the heavier types of machinery, since much of this kind of equipment is produced in quantities so small that it does not lend itself to American methods of production and calls rather for the individual skill that is found more highly developed in the principal European countries. Nevertheless, as a whole, we have at hand all that is best in design and in workmanship of that class of machinery that is particularly applicable to the automobile trades, and which may be covered by lathes up to 36 in., planing machines up to 56 in., radial drilling machines up to 5 ft., milling machines up to No. 4 and gear-cutting equipment up to 48 in.

Besides we have an unquestioned superiority in the matter of those special highly productive machines that are developments of the standard equipment mentioned above and owe their inception so largely to the mass production of the sewing-machine, typewriter and automobile industries.

The Machine-Tool Industry

Diverting for the moment to the development of the machine-tool industry, prior to the war the number of men employed in the United States in the construction of machine-tools was approximately 33,000 and the output was valued at approximately $45,000,000 per year. At the peak of production during the war over 80,000 men were employed and the output was estimated as somewhere between $400,000,000 and $500,000,000 per year. These valuations, of course, do not express accurately the number of machines produced because the cost was increased very materially during the war. But, making due allowance for the non-employment prior to the war, the overtime work during the war, and the 100-per cent addition to the price of the machinery, it is reasonable to estimate that our machine-tool productivity of today, if stressed to its maximum, would be at least two and one-half times that of 1913; and it is further to be noted that the larger part of this increase is in the field of the small and medium-size machine-tools that I have already specified. Of course, a certain amount of increase and of development of production would have come in any case, through the normal processes of time and evolution. But no one I think will argue that the demand has as yet caught up with the unusual jump in machine-tool productivity that the war caused, nor will anyone be disposed to doubt that the vast majority of our factories during that period so added to their equipment that their normal demands, with a requisite allowance for the increase in equipment needed to meet their increasing trade, have for some time past been discounted. The great majority of the larger automobile factories possess surplus equipment, the full utilization of which is not likely to occur for some time to come. Some of this equipment has been so strained and injured that it must be replaced within a much shorter period than would be the case had it been operated under peace conditions. But, even allowing for this, I think you all will find that the factories you represent possess far more equipment of the standard types than can be utilized, particularly if the peak points in the production of automobiles could be ironed out. Consequently, the machine-tool builder, who looks toward a full utilization of his plant, will use all his engineering ability to develop some new machine, the output of which shall be so great that it will relegate to the discard all the machines previously produced by him, even though they may have been so well constructed and so well used that their productive life is still a matter of several years. He will do this on the theory, of the accuracy of which his sales department will endeavor to convince you, that you cannot afford to be without the newer machine because of the marked increase in production of the newer tool. If we could buy machines solely on the increase in production, the road would be easy, but this we should not do.

In the great majority of cases, assuming that a condition of a surplus of equipment does prevail, then the measuring stick by which we shall consider these offers is not increased production but decreased cost; and the two do not always go hand-in-hand. I am not dealing with a condition in which increased production is the essential thing from the viewpoint of the factory, because I do not believe that to be the case in most factories. My whole argument is built upon a belief that most of us have carried over from the war more machine-tools than we would by this time have acquired in normal times and under normal conditions, and that our problem is to determine whether we can afford to keep these machines or can dispense with them. Of course, if we are faced with an addition to our equipment that will permit us to produce more cars per day, our problem is greatly simplified since we would have only to select the machines that show the highest productive ability and apply to them the same general rules that will be laid down for the other case.

I think the foregoing should convince us that we have an ample source of supply; that it cannot become monopolistic; that the increased facilities at the disposal of machine-tool builders and their desire to utilize those facilities will lead them to the development of newer and better machines; and that, if we can exercise some influence, these machines may in the truest sense of the word be economical from the standpoint of the user. So much then for the market and the source of supply.

Cost of Labor Per Car

The third principle to be considered is the importance of a reduction in the cost of labor per car; you will please note that I do not say reduction in the price of labor. Our industry is so unfortunate as to be one in which the cost of labor is by no means equal to the cost of material. This fact makes it difficult to iron out our production schedules so that the same number of cars shall pass through our factories day after day. The demand for cars is more or less seasonal; that demand, reflected back to the factories, gives us our dull and prosperous periods, which we can not guard against by building up a stock of cars during the dull period, because of the tremendous inventory that we would accumulate by so doing.

Consequently, our industry offers its employes a relatively intermittent employment. To keep approximately the same number of employes throughout the year is given only to a very few of the larger shops and to a greater proportion of the smaller shops. Therefore, at certain periods, the employment department is called on to supply machine operators at a time when all other automobile manufacturers are clamoring for them. The result is that skilled operators cannot be secured and we must entrust our work to help of no skill or training in the manipulation of the machine. The automobile industry has never tackled in a large way the problem of instructing help, so that an adequate supply shall always be available. It has taken its skilled help from the other machine-tool-using industries, usually paying higher wages than most other industries could afford, and has never erected the machinery to replace the natural decrease in the available number of skilled men, or reciprocated by turning over to other industries trained men to take the places of those that have been taken.

As to the wisdom of this course there can be no question, but we are facing a condition, and those of us who select the machinery must bear in mind the type of help that may operate it. Machines that call for adjusting by hand during their operation, for accurate reading of dials or indicators, for careful setting up of the work in the machine, for a complex cycle of operations involving a developed mentality, all are to be decried, for not only do such machines limit the number of operators available, but under the stress of production the amount of scrap that the machines will produce is always entirely out of proportion to that produced by simpler equipment.

THE SPECIAL MACHINE

A natural development of the above line of thought leads us to the special machine. By this I do not mean the single-purpose machine or, better still, the single-piece machine. There is a marked difference here that must not be lost sight of; and our failure as an industry to keep this difference clearly before us has led to the adoption and use of some machines that cannot be regarded as wholly satisfactory from an economic standpoint. In an enthusiastic endeavor to reduce time and to simplify operations, a number of machines have been developed that are useful for one piece only. They act as a deterrent from change in design and, generally speaking, are open to these objections.

(1) Their original cost must be great because the engineering and designing must be absorbed by the few machines that can be made on those models

(2) There is always considerable delay in producing them, so that the loss on account of the continued use of the older machine until the single-piece

machine has been developed and tested out goes far toward overcoming the difference of the cost of labor between the single-piece machine and one of more general application that could be purchased as standard

(3) Since most machine-tools have been through a long process of development, it is certain that most special machines must pass through a long experimental period before they can reach the ideal set up by their designers, this adds further to the delay in obtaining full production

(4) The risk of break-down is much greater

(5) The delay in securing parts for replacement will be greater since all such parts are likely to be special

(6) The retention of an additional machine as assurance against break-down will often run the investment into large figures

(7) The difficulty of instantly replacing an operator

(8) The likelihood that special tools and fixtures must be designed and maintained

(9) The tendency of designers to incorporate elaborate tooling set-ups into such machines cannot be overlooked

All these are points of general application which are apt to be overlooked in the enthusiasm with which one views the statement that such a machine will turn out a given piece in so many seconds less than will a machine of a standard type. Often after a single-piece machine has been installed and satisfactorily operated, after its peculiarities of operation and tools have been fully understood and an organization has been trained that is able to maintain it in a state of efficiency, there is still the ever-present danger that a change of design may render the machine of no value whatever. There are today in the second-hand salesrooms so many of these machines that are without adjustments and are made so that they can produce only one piece that we need not go farther to see that we should step with caution.

Such machines have no value when divorced from the original purpose for which they were designed. A standard machine-tool, on the contrary, has a fixed market-value that depends upon its age and condition; and this value, carried on the books, can always be regarded as an asset. A special machine is apt to be carried on the books and to be depreciated by a nominal sum each year until a time comes when it is desired to turn the machine into dollars. A marked reduction in the inventory value must then be made through the inexorable law of supply and demand. A standard machine, on the other hand, can be transferred from one department to another and from one piece to another; its operators form a class and may be advertised for and hired under a classification, after the rates have been determined according to the location; the setting-up of the machine becomes a standard operaation; the design, purchase and maintenance of the tools are all matters of routine. In the event of a break-down, though only one machine may be in use, it is possible to secure repair parts almost immediately from the builder and with a reasonable guarantee of interchangeability.

Between the single-piece machine and the standard machine-tool is the safe position. Some machine-tool builders already have recognized, and there is no doubt that others will recognize, the special needs of the automobile business. They have produced machine-tools in which the feeds and speeds cannot be changed at the will of the operator but can be changed at the will of the executive by the transposition of gears. These machines permit adjustments but only by the set-up man. They are constructed liberally along the lines of spindles, slides, gearing, pulleys and the like and preferably are over-designed for the power that they will consume. They are lubricated fully and automatically and do not require the use of the oil-can. In the hands of the operator they are only single-piece machines and as such may be designed with a reserve of power and a rigidity much in excess of the more universal type of machine because their application is not so constrained, and they can be regarded as a perpetual asset even though the model, or the detail of a model, were discarded and another took its place.

The same general line of reasoning will apply to tools and fixtures. Immense sums of money are spent in providing new tools when models are changed or improved. These sums may be and frequently are calculated, and the money is set aside to meet the expenditure. The maintenance and upkeep of the tools depend largely on their standardization, which is more difficult if single-piece machines are used, since the designer is apt to build his tools, as well as his machine, to suit the piece. If we deplore the reduction in the number of skilled machine operators, how much more should we deplore, and at the same time censure ourselves for, the reduction in the number of skilled tool and die makers.

It is true that an attempt has been made, and in some shops is well under way, to split up the tool-making and the die-making departments into various groups. But this, of course, is not applicable to the smaller shops, and at its best can only reduce the requirement and not abolish it. The tool designer is another of our operating units that each year is becoming more rare. I do not mean that we cannot get enough applications from tool designers, but I do say that a much lower percentage of capable men is to be found. The vision and the administrative capacity may be there, but the instruction or apprenticeship course that develops a high-grade machinist, a tool and die maker or a tool designer is very sadly lacking. Many of the fixtures that we apply today either to standard or to special machines bear evidence of having been made by a novice. There is a glorification of the complicated. The injunction to make two ears of corn grow where one grew before evidently has been taken literally.

If any of you have analyzed the tools and fixtures in your own shops and have compared them with simpler fixtures, not from the point of view of theory or design, but from that of practical application and of how much a part produced will cost, you will be ready to agree with me on this point. I have in mind a particular example in our factory, a certain brake connection in which a slot has to be milled to remove a binding strip that holds the two halves of a piece together during the casting process. The removing of this binding strip calls for no particular accuracy, requires no power and would be regarded as a simple operation to be accomplished on a hand milling machine with a very simple fixture; the total cost of the complete equipment would not exceed $500. Such an equipment could produce approximately 700 pieces per day. With an unskilled operator, a cheap tool equipment, no floor space and practically no tool-designing or tool-maintenance charges, two of these equipments would have taken care of all the requirements of our plant for a long time. Nevertheless, the actual equipment installed consisted of a very large rotary milling machine, upon the table of which was mounted a fixture that accommodated approximately 40 pieces, the fixture and the machine in combination costing about $6,700. One machine would, of course, take care of the requirements of the plant but, as an assurance against break-down, a duplicate equipment was ordered, so that the investment was about

$13,400, or $12,400 in excess of the first mentioned equipment. Had the second machine not been ordered and a second fixture been deemed sufficient, there would still have been an outlay of about $9,000.

It would be easy to show that the big machine with one operator, on a basis of 600 cars per day, would produce a piece more cheaply than the two machines with two operators; but this is a condition that we, and I think most of you, do not experience. We may have a production of 600 cars per day for 1, 2, 3 or 4 months but we do not have it for 12 months. We could afford to run those two small machines with two operators during our peak period, since the cheapest kind of help could be used on them, better than we could afford to spend the money that was spent for the expensive equipment. If we do the obvious thing, discard this casting and use in its place the stamping, we shall have on our hands two fixtures, one of which costs more than the full machine and fixture equipment that was considered in the first case. These fixtures have no resale value and reduce our inventory or assets by the amount of their original or depreciated value. Furthermore, such a machine, with its multiplicity of holding devices, will produce work that varies more than that which comes from the simpler machine. The floor inspector, passing from time to time, can take one of the pieces from the small machine and be sure that those that preceded it will have like accuracy. If a machine has a multiplicity of holders he cannot be so sure and the inspection charge will be increased. The scrap will be increased for the same reasons that make for a higher inspection charge. As the machine is run under the conditions of stressed production, you will find that some of the compartments are out of order and cannot be used, so the vaunted high production may be reduced, depending upon the number of compartments that are discarded. You may say that all this is bad management, that the compartment should not be permitted to get out of order, but we are talking as practical production men and we know that if a fixture at our peak period will produce three-quarters or seven-eighths of its true output, we are likely to continue in that state until a letting-up of the demand permits us to repair it.

I shall touch also upon the importance of avoiding break-downs that call for the services of skilled tool makers when such men are at a premium. If our equipment is of such a type that the average machinist can effect a satisfactory repair, we shall be that much ahead when the tight point comes and calls for immediate repair.

Skilled Workmen for Standard Machines

Another point that we must consider in the use of standard machines is the supply of skilled help that is yearly being turned out of the plants in which these machines are produced. Some of the machine-tool builders make a special point of training men, either in their own plants, or those of their customers, and of instructing them in the better handling of the machines and in the adjusting and setting-up, even to the point of effecting repairs. Such men are the nucleus around which a classification of labor is built; they call for no breaking-in and for that reason simplify the labor problem. The main reason why companies take this step is that most of the machines they produce were considered in the past to be somewhat more complicated than ordinary machines of that period and, to offset a high tool-repair or maintenance charge being made against the machines, which would of course react against their sale, they have seen fit to train satisfactory operators; of such operators we should avail ourselves thoroughly. These men should, wherever possible, be incorporated into a machine-repair gang, because it goes without saying that the more complicated the machine, the more skill and special training required to dismantle and repair it. Much harm may be done by unskillful attempts to repair a machine.

Basis of Purchase

Now, having in mind these general considerations, we come to the reasons for purchasing new equipment or new machines. The one most frequently encountered is that the new machine will save money. It is not always expressed that way; it is sometimes put that the new machine will reduce the labor cost or will turn out a piece more quickly than under the old method; but these are not the real things to be considered. The only satisfactory reason is to reduce the cost and not to reduce the labor charge or increase the production per man; and in this cost reduction appears the consideration of the items that I have already touched on. The second reason is to increase production; in other words, to turn out more parts per year or per season. In this case the consideration will be whether to put in more machines of the

Baker

112 340	1.30	$535.00
200	2.00	$535.00
312	1.40	$535.00
440	1.00	$535.00
TAL...	$9.70	$4,280.00

duction per	Cost per	Cost of New

Supplement to Selection of Machine-Tools by A. J. Baker

SAVINGS TO BE ACCOMPLISHED BY INSTALLATION OF NEW EQUIPMENT BASED ON THE PRODUCTION OF 500 MODEL 4'S PER DAY

Part No.	Name of Part	Name of Present Machinery	Name of New Machinery	Saving of Floor Space, Sq. Ft.	Increase of Floor Space, Sq. Ft.	Book Value Special Equipment To Be Discarded	Book Value of Present Machinery	Resale Value of Present Machinery	Cost of New Machinery and Tools	Required Expenditure	Saving per Car	Cars to Amortize at 500 per Day
300,562	Inner Brake-Band	2—Silver Drills	1—No. 2 Avey Drill	8	......	$564.00	$225.70	$159.00	$2,800.00	$2,641.00	$0.0060	221,700
303,584	Brake Locker Lever Bracket	4—No. 310 Baker Drills 4—25" Barnes Drills 1— 8" Henry & Wright Drill	1—No. 2 Avey Drill 1—No. 3 Avey Drill	97½	......	1,885.00	3,410.26	3,332.00	12,325.00	8,993.00	0.0470	191,240
300,387	Steering-Knuckle Arm	2—No. 310 Baker Drills 5—No. 4 Bardons & Oliver Lathes	2—Daniels Automatic Multiple-Spindle Chucker	113	......	2,475.00	7,808.35	5,981.00	18,868.00	12,887.00	0.0257	253,266
300,239	Crankshaft	6—16x32 Landis Grinders	6—Wickes Heavy-Duty Crankshaft Lathes	......	54	......	16,097.76	15,943.00	27,000.00	11,057.00	0.0773	146,357
300,180	Nut Welding ¼x28 Thread	1—Garvin Vertical Tapper	1—D12 Fox Multiple Drill and Tapper	......	......	35.00	411.57	364.00	3,233.00	2,869.00	0.0060	444,390
300,382	Front Axle	3—Espen Lucas Millers 2—25" Barnes Drills 3—No. 7 Becker Millers	1—Ingersoll Drum Type Axle Miller	243	......	3,393.00	9,860.46	10,344.00	14,386.00	4,042.00	0.0610	71,125
300,313	Piston Pin	4—10x28 Norton Grinders	2—Model B Sanford Centerless Grinders	70	......	......	9,692.84	5,826.00	5,000.00	......	0.0070	......
300,620	Brake Support Bracket	3—No. 1 Kempsmith Millers 1—No. 3 Kempsmith Miller	1—40" Ohio Rotary Tilting Miller	89	......	1,425.00	5,023.65	2,880.00	4,520.00	164.00	0.0170	120,897
305,166	Brake-Spring Bracket	2—No. 3 Kempsmith Millers 1—18" Cincinnati Miller 1—No. 7 Becker Miller	1—40" Ohio Rotary Tilting Miller	85	......	1,650.00	4,589.40	3,323.00	4,735.00	1,412.00	0.0120	134,437
300,385-6	Steering-Knuckle	4—25" Barnes Drills 1—No. 7 Becker Miller 1—No. 1 Tol. Hd. Miller	1—40" Ohio Rotary Tilting Miller	183	......	1,282.00	1,483.96	1,254.00	8,990.00	7,736.00	0.0400	202,549
300,385-6	Steering-Knuckle	2—16" American Lathes 3—3½x30 LoSwing Lathes	2—No. 9 LeBlond Multi Cut Lathes	42	......	337.50	4,679.24	3,454.00	4,564.00	1,110.00	0.0120	53,889
300,209	Camshaft	8—10x28 Norton Grinders	4—Melling Cam Turning Lathes	167	......	......	19,385.68	11,872.00	15,805.00	3,933.00	0.1020	42,761
300,675	Steering Arm	3—No. 4 Bardons & Oliver Lathes	Daniels Automatic Multiple-Spindle Chucking Machine	47½	......	318.75	4,683.00	2,511.00	10,050.00	7,539.00	0.0190	399,866
	TOTAL			1,144½	54	$13,365.25	$86,351.57	$67,243.00	$132,276.00	$65,859.00	$0.4220	156,064

PROPOSED METHOD OF MACHINING PART No. 300,239—CRANKSHAFT
Submitted by: JONES & LAMSON MACHINE CO.
Compared with our present method, figured on a basis of 500 cars per 8 hr.

Oct. 24, 1922.

PRESENT METHOD

Model 4—1 Per Car

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Present Resale Value of Machinery
Face Sides and Turn Flange Face Inside of Flange	18-In. American Lathe 18-In. American Lathe	5 2	14.0 42.5	112 340	$4.00 1.30	$535.00 $535.00
Space and Rough Turn Rear Bearing	18-In. American Lathe	3	27.5	200	2.00	$535.00
Finish Turn Flange	18-In. American Lathe	2	39.0	312	1.40	$535.00
Under Cut Flange	18-In. American Lathe	2	55.0	440	1.00	$535.00
				TOTAL...	$9.70	$7,490.00

PROPOSED METHOD

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Cost of New Machinery
Turn Flange End for Grinding	Double Carriage Fay Automatic Lathe (Flanders Type)	6	10.8	92.8	$5.20	$3,143.50
				TOTAL...	$5.20	$18,861.00

REMARKS

Total resale value of machines for present method	$7,490.00	Present labor cost per hundred	$9.70
Total cost of machines for proposed method	$18,861.00	Labor cost by proposed method	$5.20
Total______on machines for______method	$11,371.00	Saving of labor per hundred	$4.50
Total cost of new tools for proposed method		Operators eliminated by proposed method	
Total increase of expenditure (including machines, tools and		Machines figured for factor of safety	
floor space)		Cost of taking down and installing new equipment	
Tools retired are good for production of______cars per day		Cars required to pay for new equipment	252,700
Value of tools retired (original cost of tools less 50% per annum)		(including machinery, tools and installation)	

PROPOSED METHOD OF MACHINING PART No. 300,239—CRANKSHAFT
Submitted by: JONES & LAMSON MACHINE CO.
Compared with our present method, figured on a basis of 500 cars per 8 hr.

Oct. 24, 1922.

PRESENT METHOD

Model 4—1 Per Car

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Present Resale Value of Machinery
Space Gear End Bearing Rough Turn Gear End	18-In. American Lathe 18-In. American Lathe	2 2	56.0 42.5	448 340	$0.99 1.30	$535.00 $535.00
Finish Turn Gear End	18-In. American Lathe	2	55.0	280	1.58	$535.00
Neck and Chamfer Gear End	18-In. American Lathe	1	69.0	552	0.80	$535.00
Under Cut Gear End	18-In. American Lathe	1	162.0	1,296	0.39	$535.00
				TOTAL...	$5.06	$4,280.00

PROPOSED METHOD

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Cost of New Machinery
Turn Gear End for Grinding	Double Carriage Fay Automatic Lathe (Flanders Type)	3	23.6	202.9	$2.35	$2,883.50
				TOTAL...	$2.35	$8,650.50

REMARKS

Total resale value of machines for present method	$4,280.00	Present labor cost per hundred	$5.06
Total cost of machines for proposed method	$8,650.50	Labor cost by proposed method	$2.35
Total______on machines for______method	$4,370.00	Saving of labor per hundred	$2.71
Total cost of new tools for proposed method		Operators eliminated by proposed method	
Total increase of expenditure (including machines, tools and		Machines figured for factor of safety	
floor space)		Cost of taking down and installing new equipment	
Tools retired are good for production of______cars per day		Cars required to pay for new equipment	164,304
Value of tools retired (original cost of tools less 50% per annum)		(including machinery, tools and installation)	

PROPOSED METHOD OF MACHINING PART No. 300,239—CRANKSHAFT
Submitted by: JONES & LAMSON MACHINE CO.
Compared with our present method, figured on a basis of 300 cars per 8 hr.

Oct. 24 1922.

PRESENT METHOD

Model 4—1 Per Car

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Present Resale Value of Machinery
Face Sides of Flange and Turn Face Inside of Flange	18-In. American Lathe 18-In. American Lathe	3 1	14.0 42.5	112 340	$4.00 1.30	$535.00 $535.00
Space and Rough Turn Rear Bearing	18-In. American Lathe	2	27.5	200	2.00	$535.00
Finish Turn Flange	18-In. American Lathe	1	39.0	312	1.40	$535.00
Finish Cut Flange	18-In. American Lathe	1	55.0	440	1.00	$535.00
				TOTAL...	$9.70	$4,280.00

PROPOSED METHOD

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Cost of New Machinery
Turn Flange End for Grinding	Double Carriage Fay Automatic Lathe (Flanders Type)	4	10.8	92.8	$5.20	$3,143.50
				TOTAL...	$5.20	$12,574.00

REMARKS

Total resale value of machines for present method	$4,280.00	Present labor cost per hundred	$9.70
Total cost of machines for proposed method	$12,574.00	Labor cost by proposed method	$5.20
Total______on machines for______method	$8,294.00	Saving of labor per hundred	$4.50
Total cost of new tools for proposed method		Operators eliminated by proposed method	
Total increase of expenditure (including machines, tools and		Machines figured for factor of safety	
floor space)		Cost of taking down and installing new equipment	
Tools retired are good for production of______cars per day		Cars required to pay for new equipment	160,530
Value of tools retired (original cost of tools less 50% per annum)		(including machinery, tools and installation)	

PROPOSED METHOD OF MACHINING PART No. 300,239—CRANKSHAFT
Submitted by: JONES & LAMSON MACHINE CO.
Compared with our present method, figured on a basis of 300 cars per 8 hr.

Oct. 24, 1922

PRESENT METHOD

Model 4—1 Per Car

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Present Resale Value of Machinery
Space Gear End Bearing Rough Turn Gear End	18-In. American Lathe 18-In. American Lathe	1 1	56.0 42.5	448 340	$0.99 1.30	$535.00 $535.00
Finish Turn Gear End	18-In. American Lathe	1	35.0	280	1.50	$535.00
Neck and Chamfer Gear End	18-In. American Lathe	1	69.0	552	0.80	$535.00
Under Cut Gear End	18-In. American Lathe	1	162.0	1,292	0.34	$535.00
				TOTAL...	$4.93	$2,675.00

PROPOSED METHOD

Operation	Name of Machine	Machines Required	Time Study	Production per Machine in 8 hr.	Cost per 100	Cost of New Machinery
Turn Gear End for Grinding	Double Carriage Fay Automatic Lathe (Flanders Type)	2	23.6	202.9	$2.35	$2,883.50
				TOTAL...	$2.35	$5,767.50

REMARKS

Total resale value of machines for present method	$2,675.00	Present labor cost per hundred	$4.93
Total cost of machines for proposed method	$5,767.00	Labor cost by proposed method	$2.35
Total______on machines for______method	$3,092.00	Saving of labor per hundred	$2.58
Total cost of new tools for proposed method		Operators eliminated by proposed method	
Total increase of expenditure (including machines, tools and		Machines figured for factor of safety	
floor space)		Cost of taking down and installing new equipment	
Tools retired are good for production of______cars per day		Cars required to pay for new equipment	116,240
Value of tools retired (original cost of tools less 50% per annum)		(including machinery, tools and installation)	

type already in use or to purchase some machine that is an improvement but of the same general type, or to get an entirely new kind of machine.

There is much to be said for maintenance standards. If the records show that the tool you have been using is up to the average in productivity, it would be foolish to change to another make, even if a somewhat greater output could be shown. Unfortunately many machine-tools of the same general classification differ so much in detail that the equipment of one cannot be transferred to another; the T-slots in the tables, the taper hole in the spindles, the thread on the spindles, the form of the tool-holder, the method of clamping the tools, the arrangement of the control levers, all these differ very widely. You are therefore forced to make up special fixtures differing in some details from those that you have been using on other machines. This means that if a breakdown occurs, and you have planned for it and have the extra tools available, you will have had to carry just twice as many fixtures in excess of actual requirements. If you have more than one make of machine you will not have the facility of immediate interchangeability; you will not be able to transfer the operators with any degree of certainty; the foremen will spend much more time in the instruction of the men; the time-study department will have to make changes in the times, because the speeds and feeds may differ somewhat; and it may mean even an adjusting of rates. Further, you will have to keep in stock certain replacement parts for these machines that, of course, will be doubled in number if you have in use more than one type of machine.

Now, if we decide to put in an entirely new type of machine, we should give the matter a very careful analysis. The blanks that we are using at the Willys-Overland plant are shown. They can be used with additional machines, as the need of such machines appears in our schedule of production and also with new machines that are brought to our attention through the production of our competitors' shops, the trade journals or the visits of representatives of the builders or vendors of the machines. On the first blank reproduced you will notice the usual information as to the name of the part and the company by which the proposal is submitted and the basis on which the figuring is done so far as the number of cars per day is concerned. From there on a detailed comparison is made that gives the current operation as against the operation suggested, the name and number of machines required, the time study, the time per piece, the production per machine per 8 hr. and the cost per hundred.

I fear, generally speaking, that is all the information that is considered in the purchase of new equipment; we do not figure on the cost per piece or per hundred expressed in wages paid out on the job. You will note, however, that we go a little farther; we have specified the present resale value of the machinery now installed. This is to be used in replacement and wherever additional machines are installed; and against it we place the cost of the new machinery. In the tabulations at the bottom of the chart we show the total expenditure incurred, balancing the resale value of the machinery to be discarded against the expenditure required for the machinery to take its place. We add to this the cost of the new tools required for the proposed method and, if an increase in the floor space is required for the new tool, that also appears. Each foot of floor space carries a certain charge that varies with the building, and includes the items of power, light, heat, water, insurance and the like; in other words, a floor-space charge that is not an overhead charge. Below this is an item that may be considered only when we intend to change over to reduce costs, as it gives the production per day that the tools to be retired if the contemplated action is taken would be good for. This figure must be considered in connection with the actual labor cost of the parts, because an executive passing on this matter must know the contemplated production, as he would incline favorably toward the new equipment if he found that the old equipment were taxed nearly to its productive limit.

Below that appears the value of the tools retired, which is the original cost of the tools, less 50 per cent per year. In other words, if the special tools and fixtures had been used for a few months only, we should depreciate our inventory by the full value, or cost, of the equipment at a figure that would be carried on our books, and at the end of the year that equipment would be written off 50 per cent, the next year 50 per cent of the remainder, and so on.

A very material reduction thus takes place in the inventory value that avoids the piling-up of a so-called asset that really is no asset at all. Nevertheless, if no consideration is given to the inventory value of these tools, you might by carrying out the matter to a ridiculous extreme, wipe out in 1 day from the book value of the stock the whole item that is classified as small tools and fixtures and have nothing to show for it. Such action, of course, would involve you at once with the accounting department, and be very poor business. I do not wish to convey the impression that we actually write off our small tools, jigs and fixtures at the rate of 50 per cent per year, but for the purpose of figuring against contemplated installments, particularly if it is for the purpose of effecting reductions in cost and not of taking care of extensions in volume, this makes a rather satisfactory arrangement.

The next two items show the labor cost at present as against that of the proposed method, and the saving of money in labor charges. The next item, which is again an intangible one, shows the number of operators that would be eliminated by the proposed method. This is a matter in which the factory executive and the employment manager are vitally interested. Labor troubles and short labor markets will always be with us, and the larger and more unwieldy the business becomes in point of the number of employes, the more likely are we to have trouble in procuring and maintaining an adequate labor supply. The proportion of machines taken into account in determining a factor of safety and the cost of taking down and installing the new equipment are then listed, after which the gist of the whole matter is expressed in the final line "The number of cars required to pay for the new equipment." It is of no use to say that the equipment will pay for itself in 1 year or in 2 years, because time is an uncertain element. Few men are able to estimate exactly how much work will be produced by a factory in 1 or 2 years. They may give a general average, but a progressive company should climb steadily. It seems better, therefore, to say that to pay for this saving a certain number of cars will be required. This gives two avenues for criticism: (*a*) the approximate time the cars will take to absorb this expenditure can be determined at the date of consideration by our knowledge of the expected output; (*b*) the number of cars that we are likely to make before the part in question is changed and the equipment is thrown out of use. When we are considering the installation of equipment that is made necessary by an increased production, this last item is not very important, but when we are approached for the consideration of some new machinery to take the place of that for which we already have spent our money and which already has been installed, this item becomes of paramount importance.

It has been very interesting to make these comparisons between some of the oldest equipment now in use in the Willys-Overland plant and some of the latest and most uptodate equipment that is being offered. We find that, even when a great reduction in time per piece is guaranteed by the machine builders and a good resale price is allowed for the old equipment, the actual number of cars required to pay for the new equipment is such that a rather effective damper is put on many installations that otherwise look as if they should be approved and authorized at once. I think that is because we consider in the installation of machines not only the cost of labor but also the inventory value of the equipment that is already in use.

As a matter of fact, after a study of 15 pieces as manufactured on our small car, substituting for the present equipment the latest and best standard machine-tools as specially developed for the automobile industry, we find that an expenditure of $132,000 is required to effect a saving per car of approximately 42 cents. Against this we have an expected resale value of machinery now in use of not more than $25,000; and this I believe is taking an optimistic view as the book value of the machinery stands at a very much higher figure than that given. In addition, some $13,500 worth of special equipment, book value, would have to be discarded, so that we should have to produce about 179,000 cars of this model before we would be justified in throwing out what is universally regarded as old machinery to give place to what is regarded as the very latest product of the machine-tool builders' art. There are, of course, some items in which a saving can be made in from 40,000 to 50,000 cars; some of them, however, run up to nearly 500,000 cars, and I will say further that we have not included in our study some of the more elaborate equipments to amortize the expense of which would involve a production of nearly 1,000,000 cars. This is on a basis of labor-cost saving only and does not include burden saving which, though theoretically applicable, would nevertheless hardly be reflected in the cost for a very long time.

I am well aware that much exception can be taken to this line of reasoning. It does not make the easiest road for the machine-tool builder to follow, but I think the figures cannot be controverted.

SUMMARY

In summarizing, I want to make these points

(1) There is a surplus, both actual and potential, of machine-tool equipment of the standard types
(2) Machine-tool builders are devoting their thought to high-production single-purpose machines of standard types
(3) The craze for special machinery is passing.
(4) Special machinery will not always stand a financial comparison with standard machinery
(5) We are not, as an industry, facing our responsibilities in the matter of training operative help for tool and die work
(6) When considering new equipment we cannot disregard the inventory value of existing equipment and the loss that would be shown on our balance sheet if the existing equipment were converted from productive machinery into excess machinery that would have to be offered for sale
(7) The only good reason for installing new machinery, old machinery or any machinery, apart from those causes where a better quality is demanded, is to reduce the total cost of production of the complete part

PROCESSING SPLINE SHAFTS BY A NEW METHOD[1]

By James A Ford[2]

The process devised by the author was evolved to eliminate the difficulties incident to the finishing of the spline and body portions of a spline shaft, such as is used in transmission gearing, by grinding after the shaft has been hardened, and is the result of a series of experiments.

The accuracy of the finished shaft was the primary consideration and three other groups of important considerations are stated, as well as four specific difficulties that were expected to appear upon departure from former practice.

Illustrations are presented to show the tools used, and the method of using them is commented upon step by step. The shaft can be straightened to within 0.005 in. per ft. of being out of parallel with the true axis of the shaft, after the shaft has been hardened, and it is then re-centered true with the spline portion.

The general practice among manufacturers of transmissions requiring a spline shaft has been, for many years, to finish the spline and body portions of the shaft by grinding after hardening, but it has been found that this process necessitates extreme care in the obtaining and maintaining of the desired form.

Since it was desirable that these difficulties be eliminated, it was decided by the corporation of which I am a representative that the best method would be to omit the grinding operation and substitute a process that would not include these grinding troubles. Experiments along this line consequently were carried through by the methods and standardization department of our corporation, and the results of its investigation are presented briefly as follows.

The accuracy of the finished shaft is the primary thought that was borne in mind during the investigation, and the other important points considered were that the

(1) Splines must be straight, in line with the axis of the shaft, uniform in width, properly spaced and smooth on the wearing portion

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[1] Detroit Production Meeting paper.

[2] Production department, Studebaker Corporation of America, Detroit.

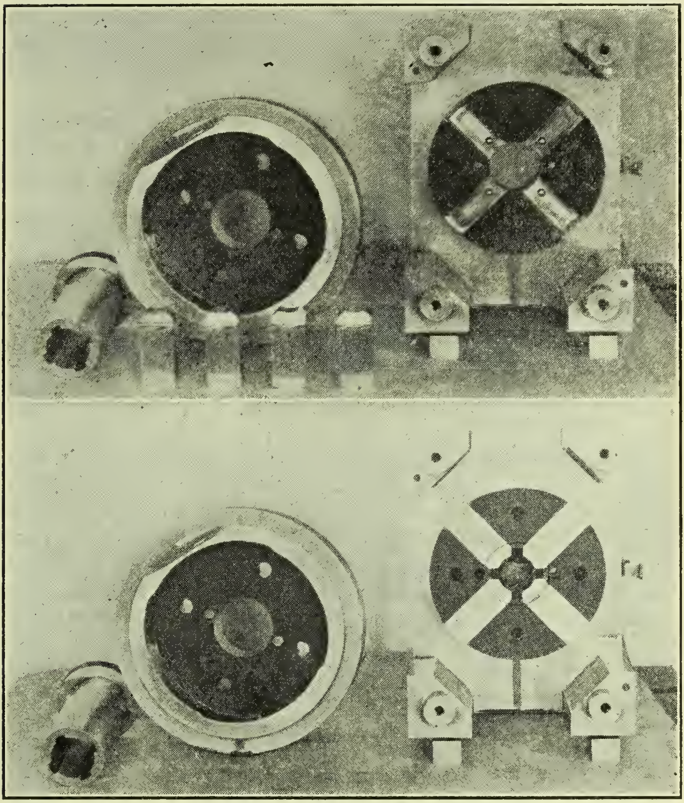

FIG. 1—TWO VIEWS OF THE DIE USED TO PRODUCE THE SPLINE SHAFTS

In the Upper View the Cutters Have Been Removed from the Die. The Cutters Are Shown in Place in the Die in the Lower View. The Cam Ring That Appears at the Left of Both Views Controls the Movement of the Cutters in Practically the Same Way That Chasers Are Controlled in a Threading Die

(2) Body of the shaft between the splines must be round and parallel with the axis of the shaft, the diameters must be held within prescribed limits at any given portion and the surface must be smooth

(3) Entire shaft must be true in relation to its axis, of the proper degree of hardness and made of the best obtainable material for the purpose, without regard to any difficulties that this requirement may introduce into machining operations

It was evident that other troubles would appear after departure from the grinding troubles that were experienced in the finishing of the shaft by the old process. The troubles that were anticipated were as follows:

(1) Difficulty in hobbing to exact dimensions, obtaining a smooth, even surface and maintaining a true form

(2) Liability of warpage in shafts during the process of hardening

(3) Variation in diameter due to hardening

(4) Variation in the width of the splines because of hardening

It was believed that a given allowance could be made for shrinkage during the hardening process and that a uniform shape and size could be determined in advance. Therefore, to overcome difficulty (1), the decision was made to draw the shaft through a die after the shaft had been carburized. It was found possible to do this by using the methods and tools that will now be described.

Tools and Methods

The die shown in the upper portion of Fig. 1 is constructed according to the same principles that apply to the automatic threading-dies now in general usage, the cutters being in the position that the die chasers ordinarily would occupy. In the lower portion of Fig. 1, the cutters are installed in the die. The cam ring is practically of the same construction as that used on a threading die, except that the ring is made stronger. This opening feature is necessary on account of having to pass the shaft back through the die, because the body of the shaft beyond the splines is larger than the body-portion between the splines and the shaft will not pass completely through the die. This is illustrated clearly by the view of the shaft shown in Fig. 2.

The shaft is entered into a bushing that is lined-up with the die. Then the shaft is pressed through the die

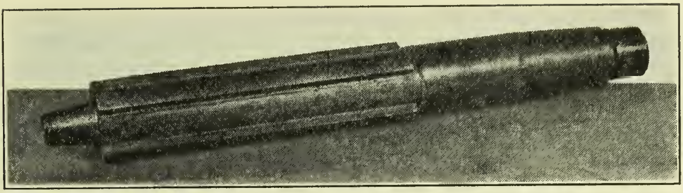

Fig. 2—A Typical Spline Shaft Showing How the Body of the Shaft Is Larger in Diameter Than the Portion between the Splines

to a stop that has been set at a sufficient distance to permit the shaft to pass through to the shaft-neck at the end of the splines, as illustrated by Fig. 3. The cutters in the die are then released and the shaft is removed.

During the experiments with this die, it was ascertained that a clearance angle on the cutter of 30 min. was about correct, and it was decided that one pass of the shaft through the die gave the most satisfactory result.

Up to this point we had a shaft that was carburized, and its body and splines were finished to the dimensions desired. After hardening a number of shafts, we were

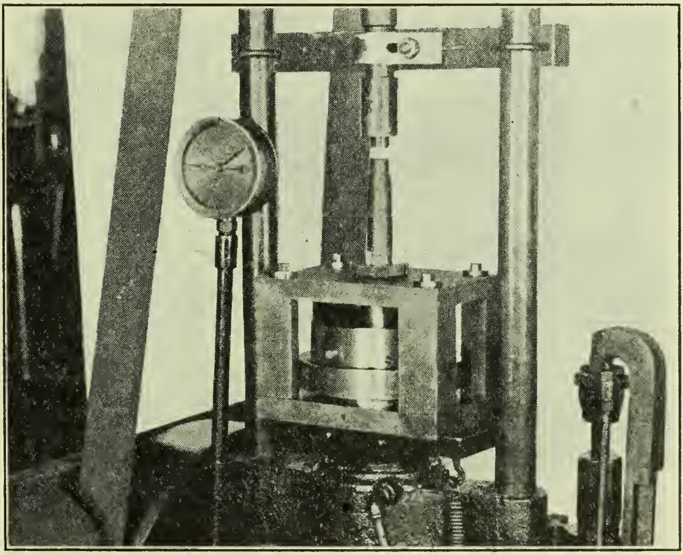

Fig. 3—A Shaft Being Pressed through the Die To Form the Splines

able to determine just what change took place in them, and an allowance was made for this in the adjustment of the die. The change was uniform to a reasonable degree in all shafts, and this enabled us to determine what the standard allowance should be.

WARPAGE

The next problem was that of overcoming the warpage in a shaft after it had been hardened. Fortunately, we found that this warpage was outside of the splined portion of the shaft.

It was very difficult to revolve the shaft on its centers and straighten it to the degree of exactness required, and the operation required too much time. However, we found it possible to straighten the shafts easily to within 0.005 in. per ft. of being out of parallel with the true axis of the shaft. Therefore, we straightened the shafts to within the 0.005-in. per ft. limit.

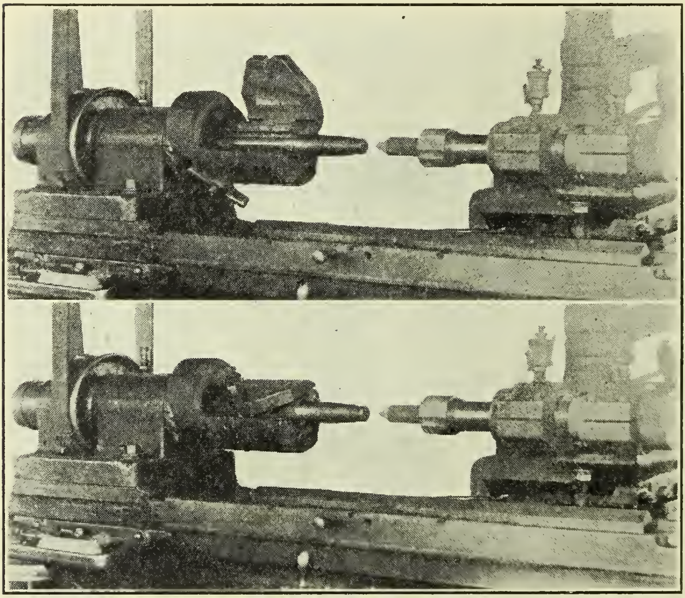

FIG. 4—GRINDING THE CENTERS IN THE SHAFT AFTER THE SPLINES HAVE BEEN FORMED

The Shaft Is Placed in the Fixture Shown in the Upper View and After the Fixture Has Been Closed and Clamped as Illustrated Underneath the Centers Are Ground with a Pencil Wheel. This Method Insures Centers That Are True with the Spline Portion of the Shaft

RE-CENTERING

After having been hardened and straightened to within the 0.005-in. per ft. limit, the shaft is gripped in a fixture mounted on the spindle of an internal-grinding machine as illustrated by the two views shown in Fig. 4. It is then revolved true with the splines while it is being re-centered by grinding against the pencil-shaped grinding-tool shown also in Fig. 4.

After the new centers have been established true with the spline portion of the shaft, the remainder of the operations follow ordinary shop practice.

FORD ENGINE-CYLINDER PRODUCTION[1]

BY P E HAGLUND[2] AND I B SCOFIELD[2]

The authors state the principles governing intensive quantity-production and describe the sources and methods of handling the basic materials that compose the Ford engine-cylinder. The fundamental plan of the River Rouge plant is outlined, illustrations being used to supplement the text that explains the reasons governing the location of the various units of the plant. Details are given of the use made of conveyors with the idea of keeping everything moving.

The relation of the blast furnace and coke ovens to the engine cylinder are commented upon, the power-house and foundry are described, and the production of the cylinder is set forth step by step.

The Ford Motor Co. began making its own cylinder-blocks in 1907, accepting the men and methods of the day. Its foundry at that time was located at Romeo, Mich., 60 to 70 miles north of Detroit. The design of the cylinder-block has changed somewhat from the one then produced, but it will suffice, for a rough comparison, in contrasting the results of the old methods with those resulting from a greatly increased production.

In 1908, we cast 50 cylinders per day; now we are producing 8000 cylinders in 16 hr. This greatly increased production is possible only with modern methods in which the conveyor plays the most important part. If you are in the least acquainted with foundry work, try to imagine what it would mean to cast 8000 cylinders on the floor by the old methods. This would seem nearly impossible, especially from the standpoint of the iron-handling problem. It would require acres of ground, the labor cost would be multiplied four or five times and working conditions would be unbearable because of the intense smoke and gas escaping from the molds. This immense foundry production represents the ultimate result of applying Mr. Ford's ideal of producing at the minimum cost.

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[1] Detroit Production Meeting paper.

[2] Production department, Ford Motor Co., Detroit.

To market cars in such unusual quantities, it was evident that automobiles had to be produced at a lower figure. This could not be done by lowering the men's wages and driving them, but had to be done by improving manufacturing methods. With this idea in mind, every economy was effected. First, the work and the material were brought directly to the men, thus minimizing wasteful transportation of material from one part of the shop to another. This was a severe blow at high labor cost, which is the principal item in manufacturing cost.

The next step, and the most difficult one, was to decrease the cost of the materials used. It demanded that the manufacturer have control over the raw materials from nature's source of supply to the finished product. Although this could not be accomplished at once, progress has been made in the right direction.

Basic Materials

The natural resources from which the automobile is derived are principally iron ore and coal. The engine cylinder is roughly 95.0 per cent iron and 3.5 per cent carbon from the coal. The remaining elements of its composition are also derived from the ore, with the exception of possibly some sulphur from the coke used.

As yet, the Ford Company does not mine all its iron ore, but a good portion of it comes from the Ford-Imperial mine, in the upper peninsula of Michigan. It is loaded into railroad cars, transported to a Lake port and conveyed directly to the blast furnace by a water route, the most economical means of transportation at present. The coal is mined in the Ford mines in Kentucky. It is then transported by rail and boat, or direct by rail, to the coke ovens and blast furnace at the River Rouge plant.

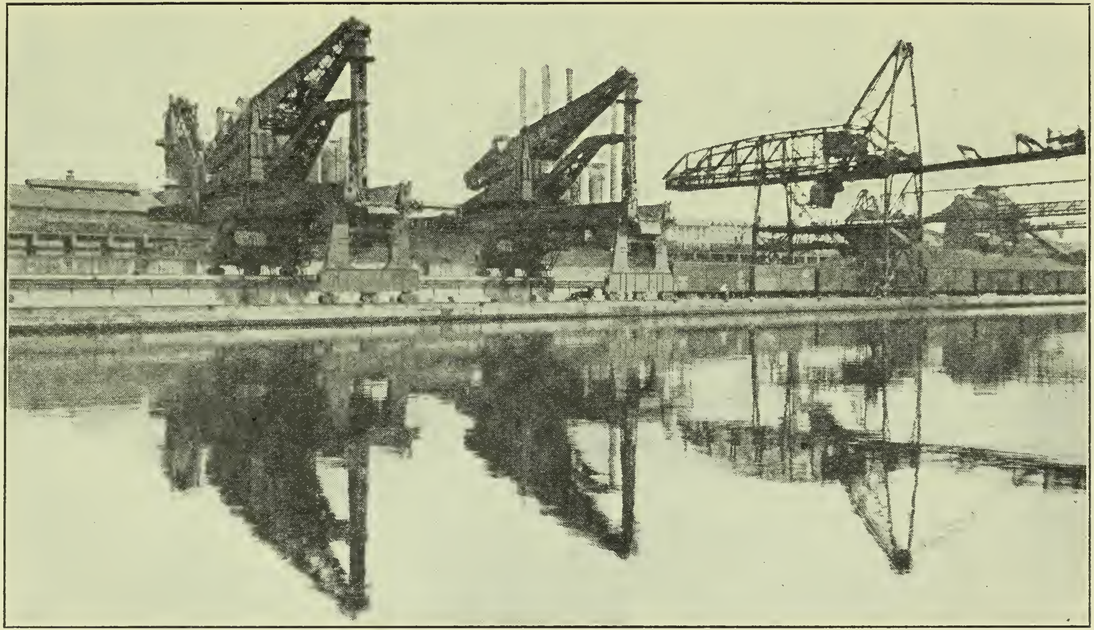

Fig. 1—Dock at Which the Ore, Coal and Limestone Required in the Production of the Ford Engine Cylinders Are Unloaded

Fig. 1 shows the unloading docks to which the ore, coal and limestone-laden boats are moored upon arrival at the River Rouge plant. Two Hulett ore-unloaders are shown at the left, and a Mead-Morrison coal-unloader at the right. These huge machines transfer the raw materials direct to the storage bins behind them. The bins, illustrated in Fig. 2, have a storage capacity of 2,000,000 tons, and are traversed by two transfer bridges each 550 ft. long. These transfer bridges, one of which is shown clearly in Fig. 2, transport to the various plant units coal, ore and limestone as they are needed.

Both ore and coal are brought to their destination with the minimum amount of loading, reloading or handling. This system also effects a considerable economy by eliminating the profits of dealers, brokers and middlemen, one of our great present-day economic wastes.

Blast Furnace and Coke Ovens

It might be well at this point to describe briefly the fundamental or general plan upon which the layout of this immense plant is based, indicating the results attained by the proper grouping of the units. The basic idea is to connect, with conveyors, each of the important units of the manufacturing plant, thus eliminating rail or truck transportation from one unit to another. To do this the units had to be built as close together as possible.

Fig. 3 shows clearly the relation of the units described in this paper. Iron ore and limestone are transferred to the blast-furnace skip-cars at *A*, where they are weighed and loaded directly into the furnace. Coal is transferred from the storage bins to the hoppers *B* and thence by mechanical conveyor through the breaker and mixer buildings to the coke-oven coal-bin. Coal arriving by rail is unloaded by the car dumper at *C* and conveyed through the same breaker and mixer buildings to the coke-oven coal-bins. The coke is mechanically conveyed from the coke bin *D* to the coke and screen station, where it is graded and distributed by mechanical conveyor to the blast furnace, to the foundry and to storage. The finer coal is pulverized in building *E* and piped direct to the powerhouse.

The powerhouse is located in the center of all of the important units; namely, the coke ovens, blast furnaces and foundry. This is done for the double purpose of placing the boilers close to the sources of by-product heat-energy, and locating the electric generators near the

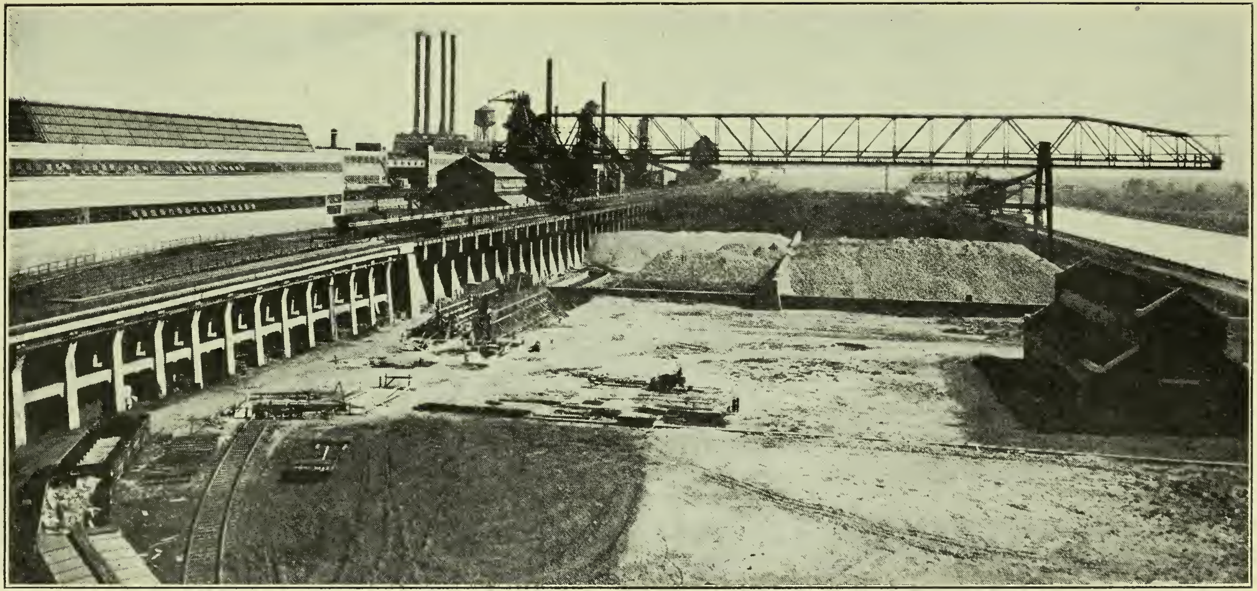

Fig. 2—The Raw-Material Storage Bins Have a Capacity of 2,000,000 Tons and Are Served by Two 550-Ft. Transfer Bridges

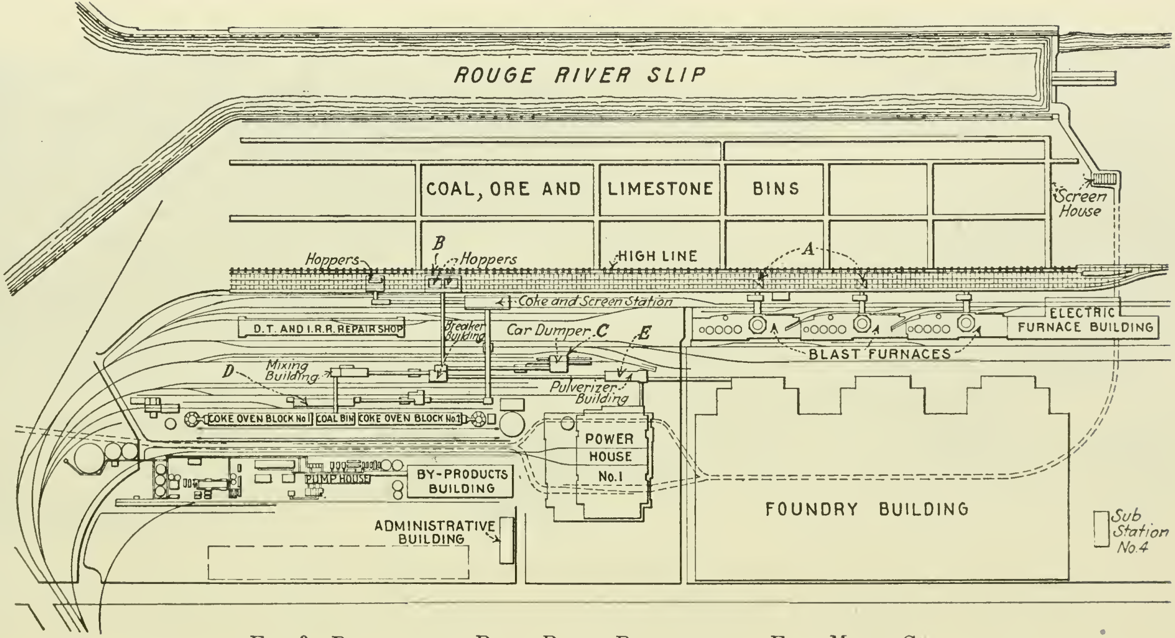
ROUGE RIVER SLIP
COAL, ORE AND LIMESTONE BINS
Screen House
B
Hoppers
Hoppers
HIGH LINE
A
Coke and Screen Station
D.T. AND I.R.R. REPAIR SHOP
Breaker Building
Car Dumper
C
E
ELECTRIC FURNACE BUILDING
BLAST FURNACES
D
Mixing Building
Pulverizer Building
COKE OVEN BLOCK No. 1
COAL BIN
COKE OVEN BLOCK No. 1
POWER HOUSE No. 1
BY-PRODUCTS BUILDING
PUMP HOUSE
ADMINISTRATIVE BUILDING
FOUNDRY BUILDING
Sub Station No. 4

FIG. 3—PLAN OF THE RIVER ROUGE PLANT OF THE FORD MOTOR CO.

units consuming electric current. The arrangement of the buildings around the powerhouse will be seen to be justified by the eventual production of castings at the minimum cost.

To obtain coke of the proper size and quality at the lowest cost, transportation expense had first to be decreased and the by-products of the coal and blast furnace, which are wasted ordinarily, had to be utilized. Coke is necessary to run the blast furnace and also the foundry cupolas. Since it usually costs the same or slightly more than coal, we buy the coal instead and reap the benefit of the by-products. Fig. 4 shows in diagrammatic form the disposition of the by-products. One ton of coal will produce approximately 0.75 ton of coke, 10,500 cu. ft. of gas, 8.5 gal. of tar, 25 to 30 lb. of ammonium sulphate and 2 gal. of light oil for the manufacture of motor fuel. From this the relative value of 1 ton of coal and 1 ton of coke is readily seen.

Of the gas lowest in by-products and heat value, 44 per cent is burned under the coke ovens; the remaining 56 per cent is used in the heating furnaces around the Ford plants in the Detroit district, any surplus during light-load periods being turned into the Detroit city mains. The tar is used in heat-treating furnaces and under the

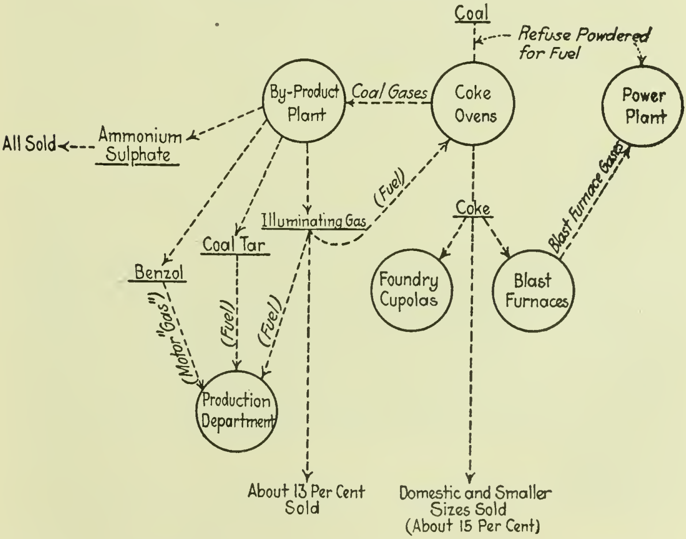
Coal
Refuse Powdered for Fuel
By-Product Plant
Coal Gases
Coke Ovens
Power Plant
All Sold
Ammonium Sulphate
(Fuel)
Illuminating Gas
Coke
Blast Furnace Gas
Coal Tar
Benzol
Foundry Cupolas
Blast Furnaces
(Motor "Gas")
(Fuel)
(Fuel)
Production Department
About 13 Per Cent Sold
Domestic and Smaller Sizes Sold (About 15 Per Cent)

FIG. 4—DIAGRAM SHOWING HOW COMPLETELY COAL AND ITS BY-PRODUCTS ARE UTILIZED

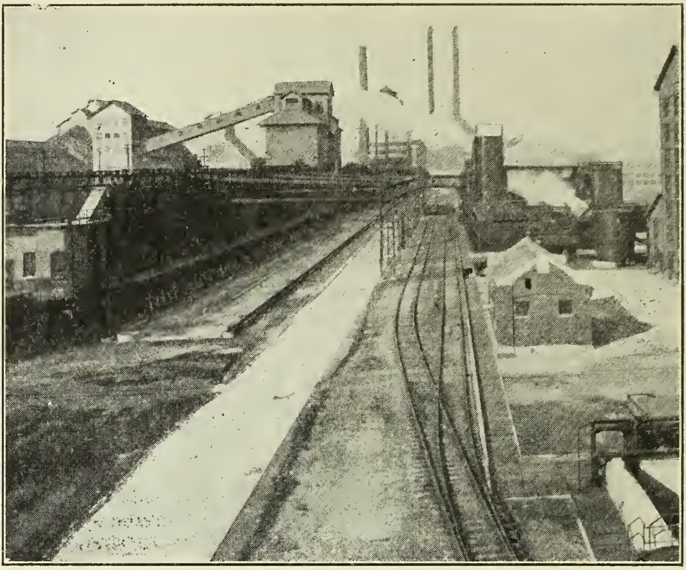

FIG. 5—A GENERAL VIEW OF THE COKE OVENS AND THE CHEMICAL PLANT

boilers in the powerhouse. Ammonium sulphate is sold as fertilizer, and the light oil is mixed with gasoline and sold as motor benzol, 10,000 gal. of benzol being produced daily. The economy effected by operating one's own coke ovens is readily seen. At present there are 120 Semet-Solvay coke-ovens in operation; they use about 2000 tons of coal per day.

A general view of the coke-ovens and chemical plant is given in Fig. 5. The ovens themselves will be noted at the left with the coal-storage bin in the center, an inclined conveyor rising into its peak from the mixing building. The benzol plant is shown at the right. Fig. 6 is a view taken on top of the coke ovens and shows the electric charging-car that transfers a load of 16 tons of pulverized coal into the four openings at the top of each oven as it requires filling. The riser pipes seen at the left collect the gases from each oven and feed them into the mains.

The side of one block of coke-ovens is illustrated in Fig. 7. The gas mains run along the top of the ovens and thence across the roadway to the pump-house and the by-products plant. The car shown in the center of the picture travels parallel to the ovens and carries a large plunger on the inner end of a deep I-beam. This plunger is pushed through each oven when the coking operation is completed, and the incandescent coke is discharged into an electric hopper-car on the opposite side of the oven. The hot coke is quenched and dumped into the coke bins, from which it is carried by mechanical conveyors to the coke and screen station for screening into blast-furnace, foundry and domestic sizes and distribution to other parts of the plant.

Fig. 8 shows one of the blast furnaces, its four air-stoves and the high-line or railroad trestle from which all material is fed into the skip-car that, in turn, travels up the inclined skip-bridge to the charging bell at the top of the furnace. There are two furnaces in the present equipment, each having a capacity of 500 tons of metal per 24 hr.

We will consider next the benefits of the situation of the blast furnaces. Most people consider a blast furnace

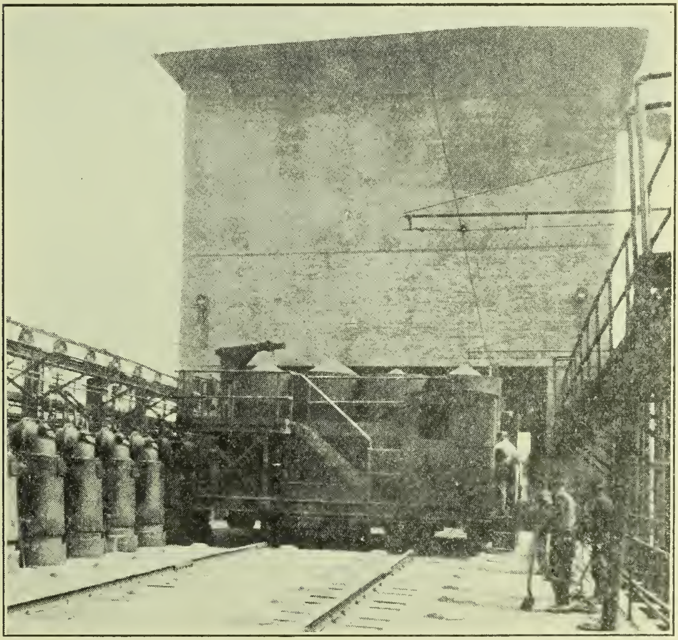

Fig. 6—View Taken on Top of the Coke Ovens Showing the Electric Charging-Car

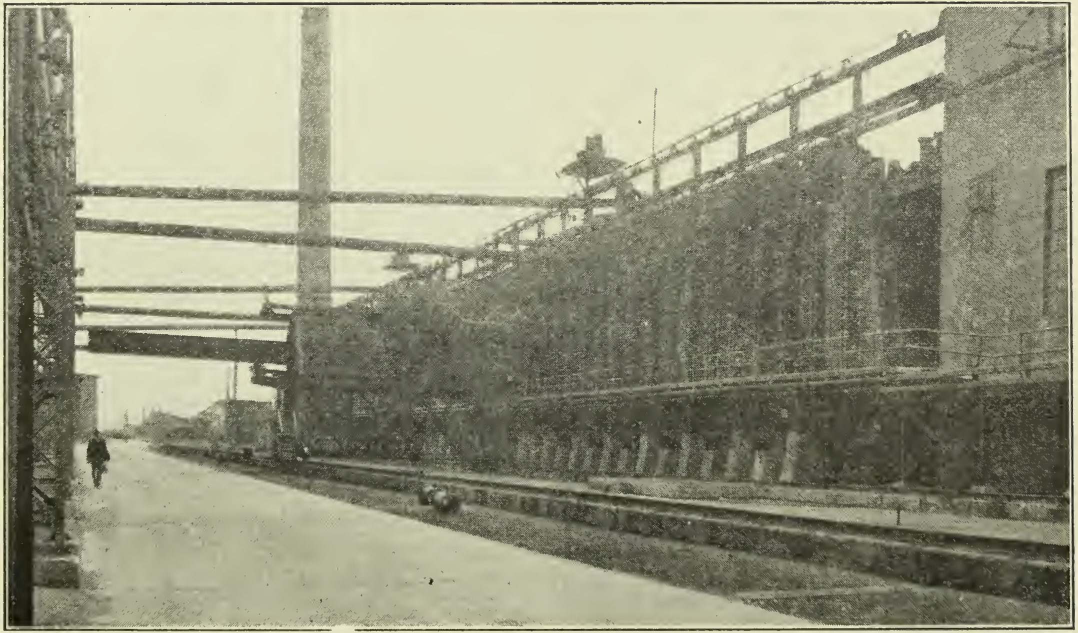

FIG. 7—THE SIDE OF A BATTERY OF COKE OVENS

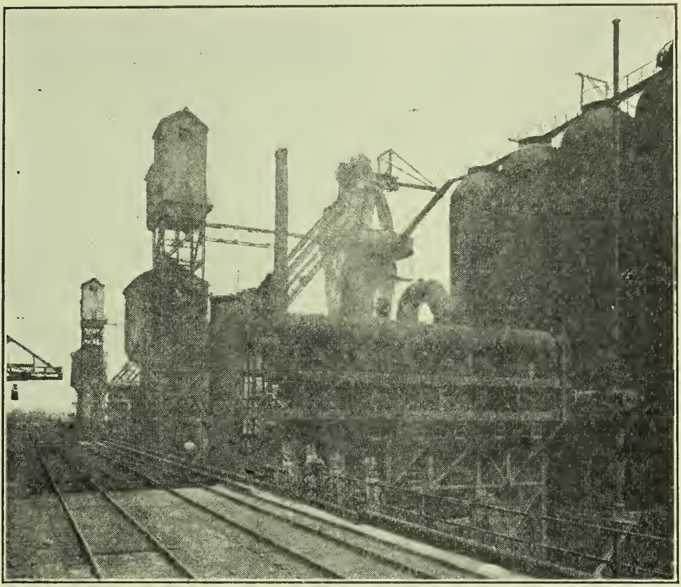

Fig. 8—One of the Blast Furnaces That Produces 500 Tons of Iron Daily

an instrument for producing pig iron only. Although its principal product is iron, its by-products are none the less valuable. Note particularly what is produced from a given charge:

Charge	*Products*
1 ton of coke	1 ton of iron
2 tons of ore	½ ton of slag
½ ton of limestone	6 tons of gas
4 tons of air	

The slag, which has been wasted, will be used very soon in the manufacture of cement, much of which will be consumed at the Ford plants for construction purposes. The gas, although of low heat-value, is a very valuable by-product. Nearly half of the blast-furnace gas is required for the four air-stoves which heat in turn the air blown into the blast-furnace. A part is burned to supply the power needed for operating the blowing engines and producing the electric power essential to the working of furnace skips, transfer cars, cranes and the like. Fifty tons of powdered coal and 36,000,000 cu. ft. of gas are burned daily under the boilers in the powerhouse, gener-

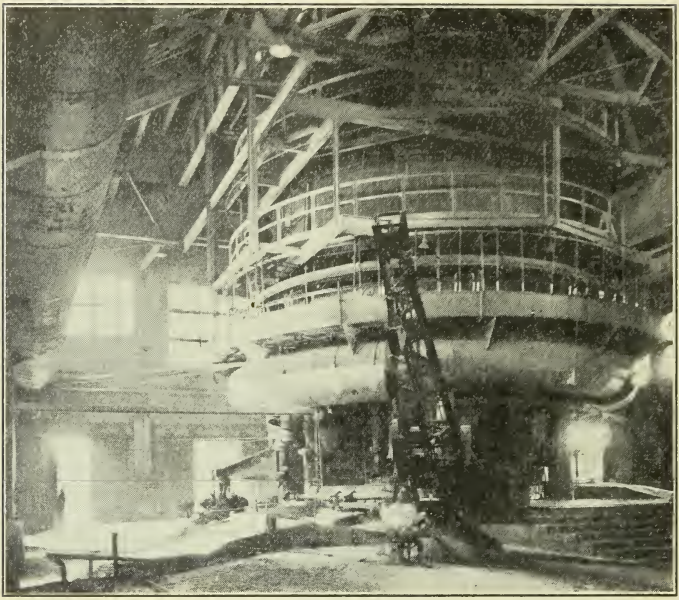

Fig. 9—Base of a Blast Furnace Showing the Stream of Molten Iron Flowing to the Ladle That Conveys It to the Foundry

ating power for departments of the plant other than the blast furnace. This shows clearly the advantage of locating the powerhouse very near the blast furnace.

Powerhouse and Foundry

The main powerhouse at the Rouge plant will eventually be the central power-source for all the Ford industries in the Detroit district. Its location at the source of large volumes of combustible by-products guarantees the generation of electric current at the minimum cost. The present equipment consists of four Ladd boilers rated at 2600 hp. each. They are fitted with a combustion system that injects liquid, gaseous or powdered fuel by air pressure. The amount of fuel is controlled electrically to meet the boiler load. Combustion is practically perfect, without ash, cinders or smoke. The customary dirt and disorder of the average boiler-room form a remarkable contrast with the bright tiled floor and orderliness of this plant.

The electric generating equipment consists of two 12,500-kw. turbo-generator units. This capacity will soon be increased. Three turbo-compressors supply the blast for the blast-furnace line, their combined capacity reaching 45,000 cu. ft. of air per min.

The location of the foundry adjacent to the blast furnace is another basic consideration in the general scheme. Up to this time, pig iron has been taken from the furnace and cast into so-called pigs. These pigs are supplied to foundries and melted in cupolas. It was at this point, in the whole process of iron ore to finished cylinder, that a considerable saving was seen to be possible. One-half of the heat generated in the blast furnace is retained in the molten iron itself. Mr. Ford believed this heat should be conserved. In a foundry where approximately 1400 tons of iron is melted per day, this item of heat loss is a considerable one.

Through misinformation or misunderstanding, many writers have stated repeatedly that the blast-furnace iron is cast directly into molds without any additions or special treatment. This is possible; but it is not practicable because the iron produced by the modern blast-furnace

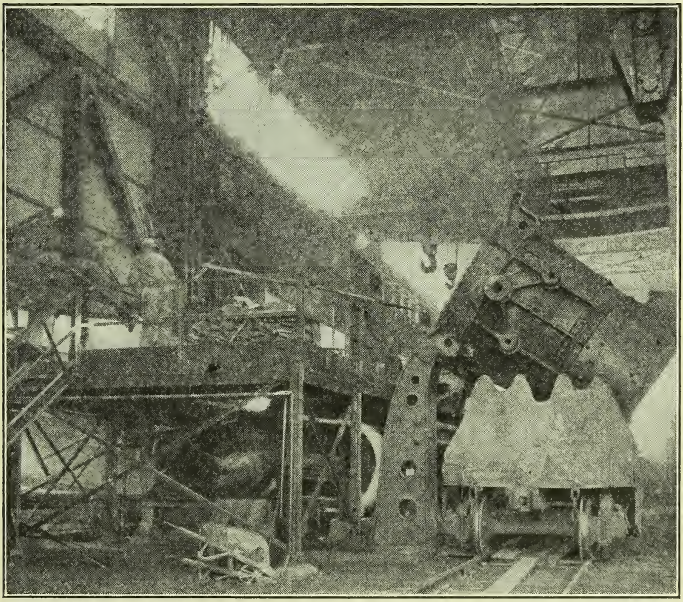

Fig. 10—The 80-Ton Ladle That Transfers the Molten Iron from the Blast Furnaces to the Foundry Discharging Its Contents into a Foundry Ladle

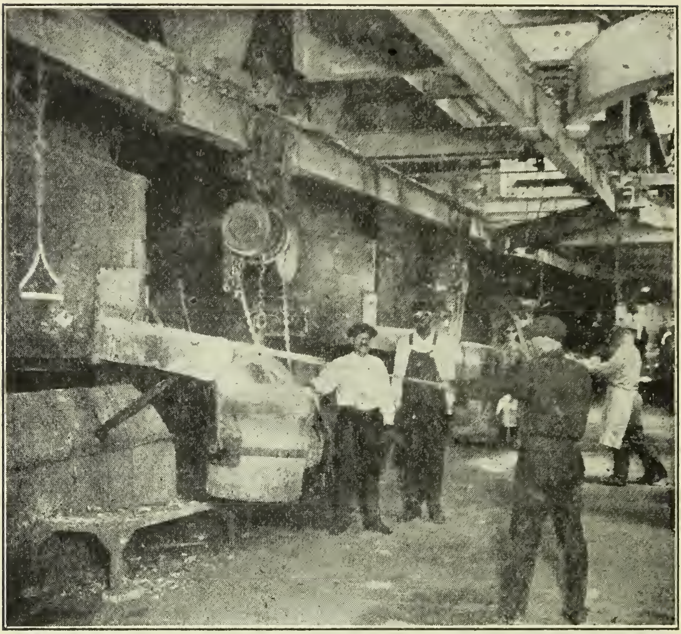

FIG. 11—POURING MOLTEN IRON FROM A CUPOLA INTO A FOUNDRY LADLE THAT IS SUSPENDED FROM AN OVERHEAD MONORAIL SYSTEM

contains too much carbon and its analysis is too variable for direct casting. To overcome this variation and to lower the carbon-content, a process has been adopted in the Ford foundry. This process enables us to use the foundry scrap and so-called back-stock that amounts to approximately 50 per cent of the iron poured into the molds. About 30 per cent of the iron poured into the Ford cylinder-mold is blast-furnace iron. The remainder is tapped from the cupolas and maintained at an analysis suitable for bringing the iron in the casting to a composition that agrees with our requirements. The percentage of manganese, sulphur and phosphorus is practically the same in both cupola and blast-furnace iron, silicon being the only element manipulated to make either soft or hard iron according to requirements. If an iron of 2-per cent silicon is required, for example, and the blast furnace is producing iron of 4-per cent silicon-content, the cupola would be run to produce 1-per cent silicon iron to give the desired chemical analysis.

Fig. 9 shows the base of a blast furnace from which the metal is being drawn. The molten iron flows through a channel directly into an 80-ton ladle outside the building. The blast-furnace iron is conveyed in this ladle by rail directly to the foundry. Here it is tilted to the position shown in Fig. 10 by a gantry crane and discharges its contents into a cylindrical container mounted on an electrically propelled car. This car takes the iron to a point between two batteries of cupolas of four each and turns its contents into the foundry ladles hung from a monorail running above the cupola spouts and at right angles to them. The foundry ladles, after receiving a weighed amount of furnace iron, are taken to the cupolas to get their portion of cupola iron and then taken to the molds to be poured. Fig. 11 shows the foundry cupolas, ladle and monorail carriers. Twenty-four cupolas are arranged in three batteries and, with three stations for furnace iron, take care of the entire production.

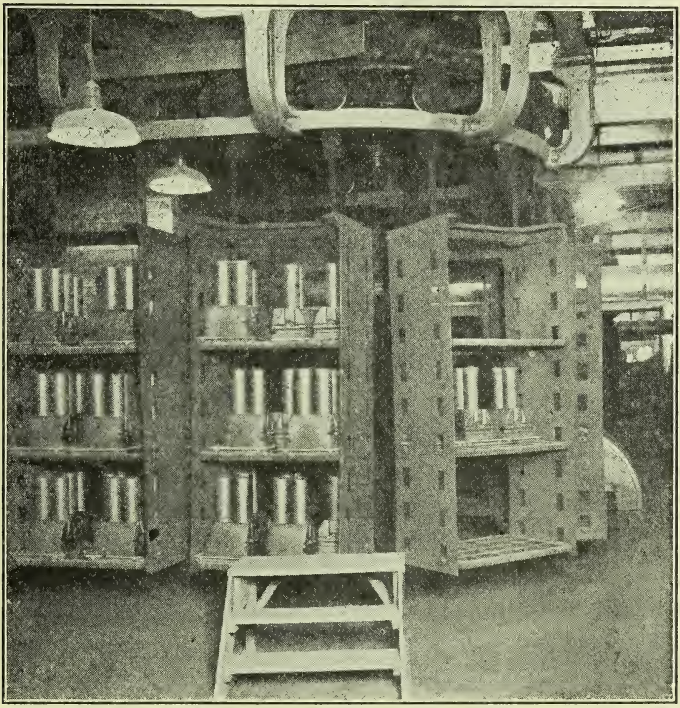

Fig. 12—All of the Material Used in the Coreroom Is Handled Progressively by Mechanical Conveyors

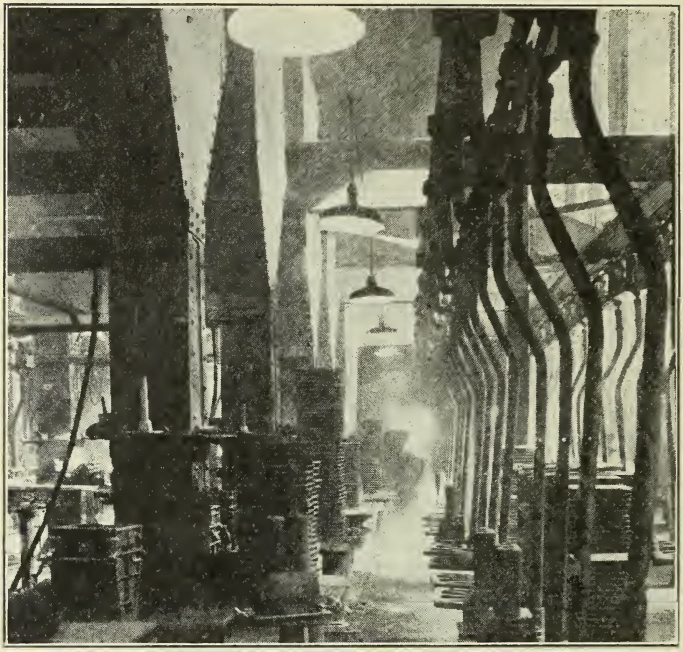

FIG. 13—ONE OF THE CONVEYORS EMPLOYED IN THE COREROOM

MAKING THE CYLINDER BLOCK

The foregoing is intended to show the magnitude and efficiency of the system needed to bring the iron to the molds at a low cost. Having provided good iron at a minimum cost, additional economies must be effected in the molding end. Heretofore, castings have been made either in molds on the floor or on benches, from which they were transferred to the floor to be poured-off. This system was crude and wasteful in addition to being laborious. The molder was compelled to carry flasks from a pile or from the yard, fit his own cores and pour the iron. In the Ford foundry every workman has all the materials he works with brought directly to him.

The system or group of conveyors, molding machines and the like comprising one unit occupies a space only 50 x 300 ft. in area. Corerooms are situated between each two systems on the same floor-level. The shake-out, where the casting is removed from the mold, is at the rear of each system. Back of this is a large cleaning-room receiving the castings from the entire system.

The cores are made from new or green sand bonded with linseed oil or a mixture containing linseed oil, or with burnt or used sand bonded with compounds made principally from pitch. The green-sand cores are used where they are surrounded by metal, and the burnt-sand cores where they are only partly surrounded. All material in the corerooms is handled progressively by conveyors as shown in Figs. 12 and 13. The core-drying oven is placed near the center of the coreroom. Cores are made at one end of the room, dried in a continuous oven and carried by conveyor from the oven to the storage space at the opposite end of the room. The cores are taken from storage to the molding machines and used as required.

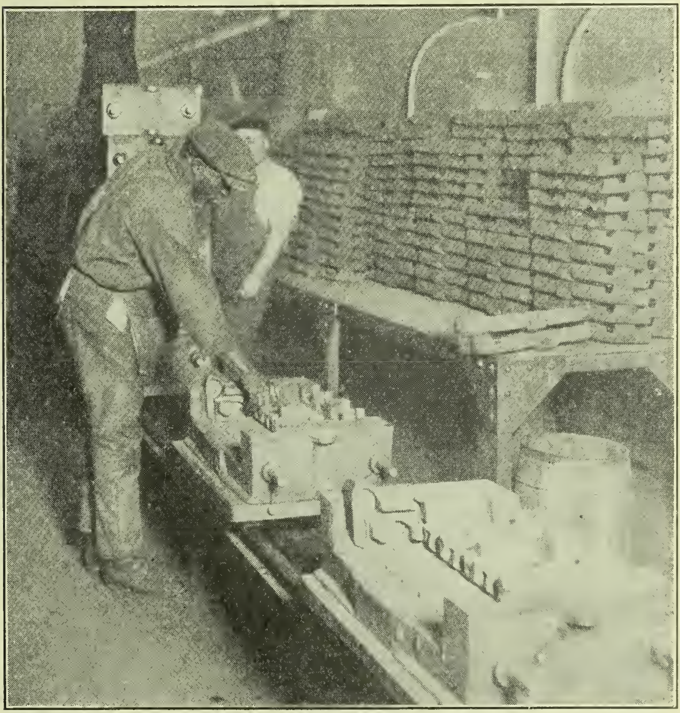

Fig. 14—Setting the Cores in the Drag Portion of a Mold as It Passes Along on a Mechanical Conveyor To Receive the Cope Portion

The cylinder is molded on the two outer chains of a system of three parallel conveyors. The two outside conveyors travel in the direction of the cupolas, while the center one moves in the opposite direction to the cleaning-room. Molding machines are placed on both sides of this system. The drag or bottom portion of the mold is rammed by hand near the starting point of the outer conveyor. Immediately after the drag has been rammed-up it is placed on the outside conveyor and, while in motion toward the cupola end, the cores are set. Fig. 14 illustrates this stage of the molding operation. Farther on the cope or top half of the mold is rammed-up and placed on the drag. After this the copes and drags are clamped together and the runner or basin that receives the molten metal is made preparatory to pouring.

Pouring the molds takes place at the cupola ends of the outside conveyors as illustrated in Figs. 15 and 16. After being poured, the molds travel to the end of the outside

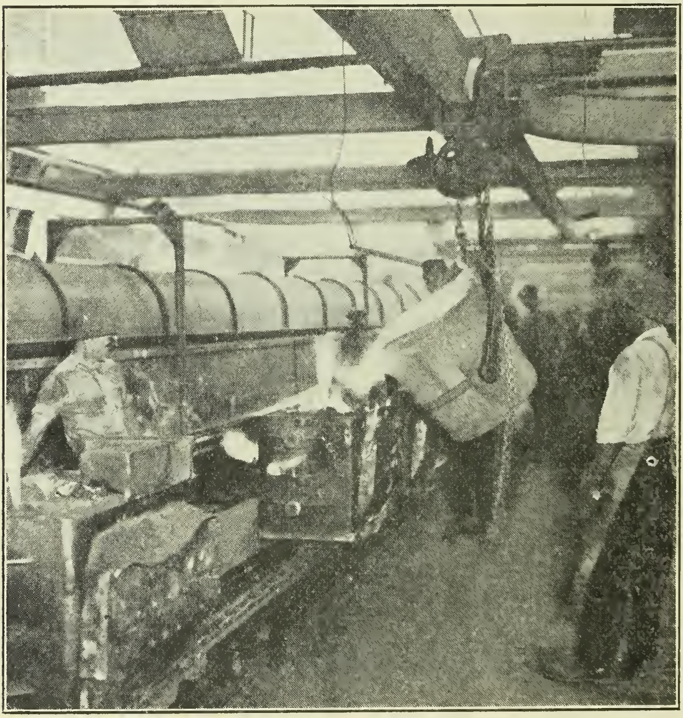

Fig. 15—Pouring the Molds

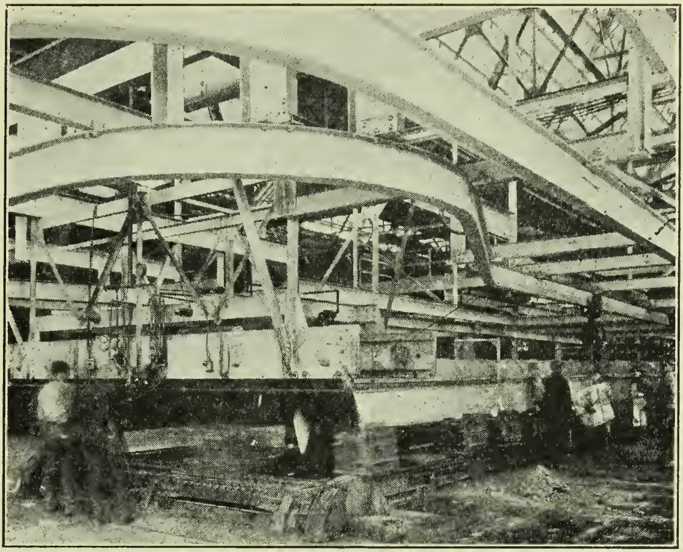

FIG. 16—AFTER THE MOLDS HAVE BEEN POURED THEY TRAVEL TO THE END OF THE OUTSIDE CONVEYOR WHERE THEY ARE SHIFTED MECHANICALLY TO THE CENTER CONVEYOR THAT CARRIES THEM TO THE SHAKE-OUT ROOM

conveyor and are shifted mechanically on a series of rollers to the center conveyor and started back toward the shake-out room. The iron solidifies and the castings become partly cool on their passage through the ventilated tunnel over the center conveyor. This removes a large amount of gas and smoke from the foundry. After the shake-out, the castings are transferred to a conveyor leading to a mezzanine floor and the loose sand drops through a grating to a sunken conveyor, as shown in Fig. 17. Enough new sand is added to the portion passing through this grating to replace the sand still adhering to the casting. The replenished sand is then automatically conveyed through a riddle or beater, to break up the lumps and also through a mixer. The whole process is mechanical except the tempering, which is looked after by a man whose duty it is to add the required amount of water to dampen the sand. After this preparation, the sand is elevated by an apron conveyor, shown in Fig. 18, which passes above the supply hoppers directly over the molding machines. Sand is drawn from these hoppers, which are shown in Fig. 19, through hand-controlled gates as each

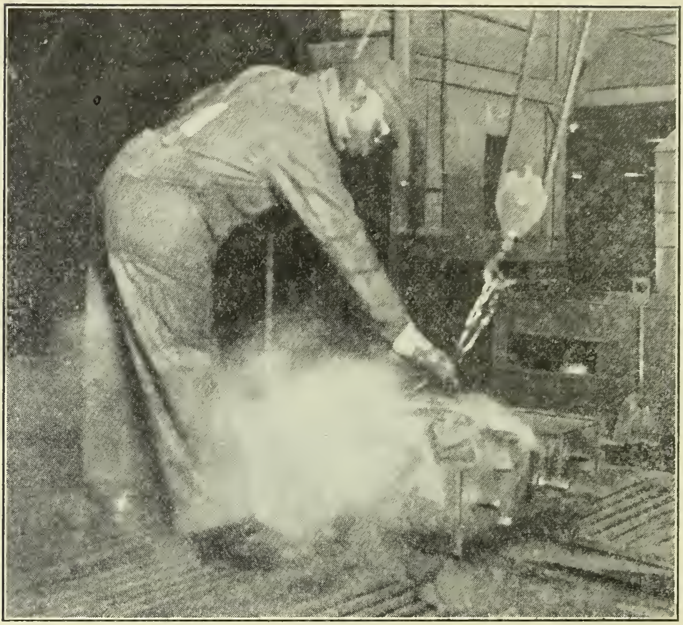

FIG. 17—AFTER THE MOLDS ARE SHAKEN OUT, THE LOOSE SAND FALLS THROUGH A GRATING TO A SUNKEN CONVEYOR

molder requires it. The sand has now completed a cycle and is ready for another mold. It is still warm from the previous casting.

The cylinder castings lie on trays until cold enough to handle and are then taken to the so-called knock-out where the bulk of the core sand is removed. This is accomplished at present by using pneumatic chisels, the cylinders resting on a grating similar to that on which the molds are shaken out. In this case, however, the burnt sand, when recovered, is used by the coreroom after proper tempering and preparation.

Cold cylinders, free from the bulk of sand in which they were cast, are then loaded on a conveyor which transfers them from the mezzanine to the ground floor where the tumbling-mills are located. A group of these mills is provided for each system. This particular conveyor also passes the cylinders between the tumblers that are grouped in two parallel rows. The cylinders are transferred by hand into the tumblers. The tumbling requires from 2 to 3 hr. The castings are removed from the tumblers by rope-hoists, returned to the same conveyor and transported to roller conveyors where the core wires are removed. From the roller conveyors, they are started on the last operations; these are performed on a slat conveyor. A crew of men on either side of this conveyor chip, grind and otherwise clean and prepare the cylinder-blocks for inspection. The rejected cylinders continue on the slat conveyor to a point farther on, where they are checked to determine the cause of rejection, removed to a truck and taken to the cupola-charging platform. The accepted castings are transferred to another slat-conveyor that runs at right angles to the cleaning conveyor; thence they pass into the machine-shop. Machining commences without delay, since the machine-shop is in the building that houses the foundry.

The Ford cylinder, although simple as compared to

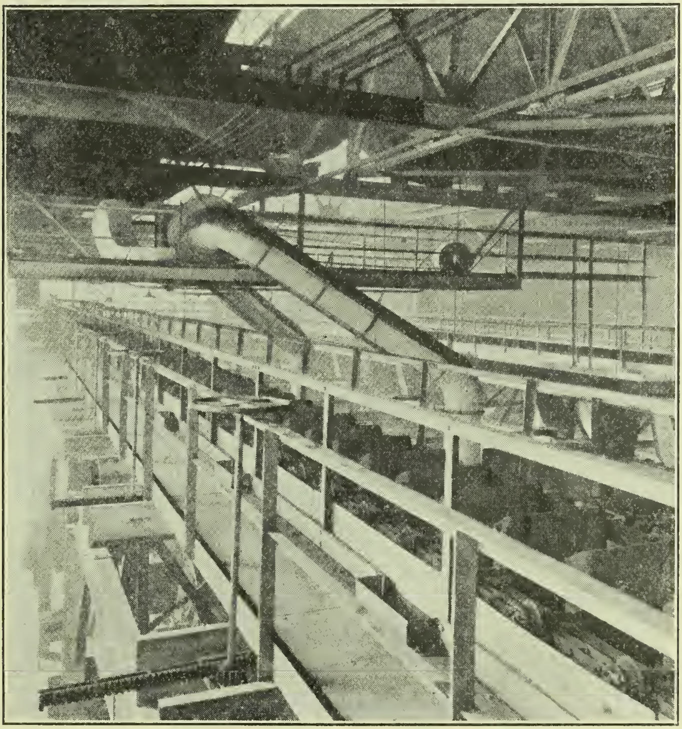

Fig. 18—Apron Conveyor That Delivers Tempered Sand to the Hoppers Directly above the Molding Machines

other cylinders, is the most complex and difficult casting in the car. In its molding many troubles come up, which are ascribable mostly to its high production. Patterns wear rapidly due to abrasion by the sand and rough usage by machine-molders. They are being repaired continually. The molding-machines also require attention nearly every week. The big problem of cylinder production lies in the constituents and usage of the sand and the iron. In a machine-shop most operations and materials are visible. The same thing is true of the patterns, machines and conveyors. A good mechanic who is always on the job is all that is required, but in the case of both the iron and the sand slight variations are hardly noticeable. Nature is not dependable when it comes to uniformity. The sand used in both molding and core-making is ever-changing. What is right one day as regards mixtures may be altogether wrong the day following. Only certain grades of sand can be utilized successfully on a job where production is high and only

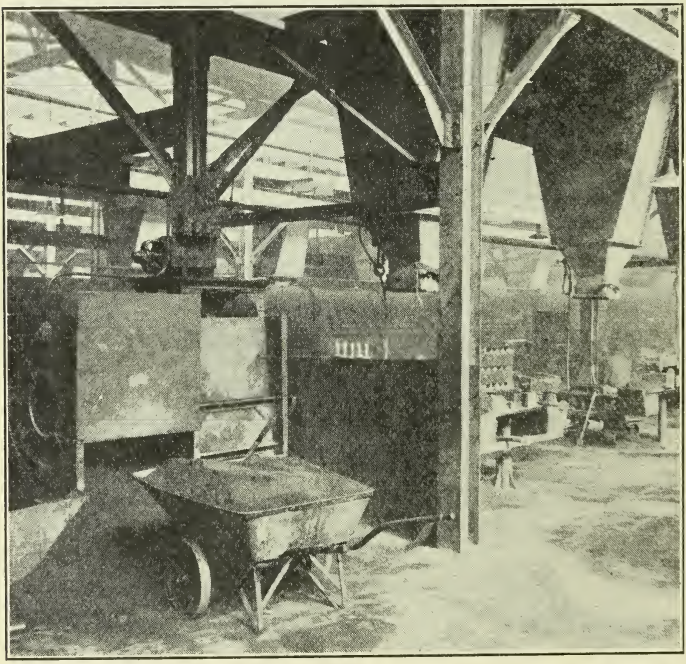

FIG. 19—HOPPERS SUPPLYING SAND THAT STILL RETAINS SOME HEAT FROM THE PREVIOUS CASTING TO THE MOLDING MACHINES

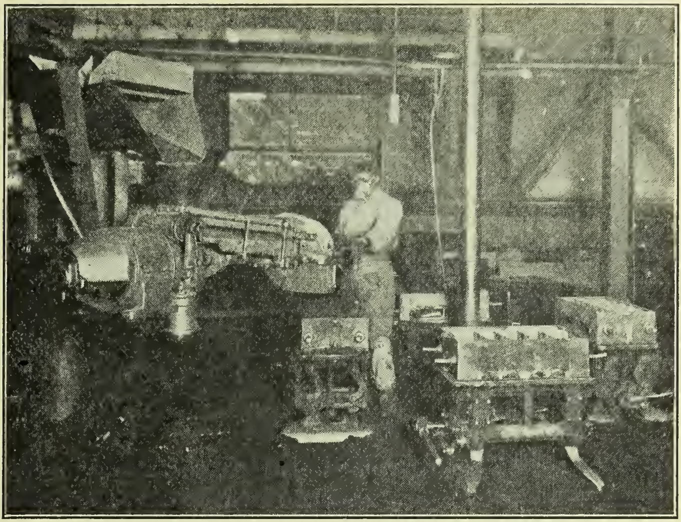

FIG. 20—AN EXPERIMENTAL MOLDING MACHINE THAT IS DESIGNED TO ELIMINATE HAND RAMMING IN WHICH THE SAND IS PACKED BY BEING THROWN INTO THE MOLD AT A HIGH VELOCITY FROM THE PERIPHERY OF A REVOLVING WHEEL

common labor is employed. Sand varies with every different source of supply and even in the same pit. Shipments of sand are watched very closely. A certain amount of bond and a grain size have been determined upon and are adhered to as closely as possible. The sand must also be rammed properly. A mold rammed too hard or not enough will produce an inferior casting. Cores must be maintained at a certain composition; the voids between the grains must be sufficient to allow free passage of gases. The cores have to be strong enough to withstand rough handling and must not be easily destructible by the high temperatures of molten or very hot iron. No set rule is applied to our sand problems; the make-up of a core is modified to meet the difficulties as they arise.

Iron for the cylinder also requires constant watchfulness. The composition of scrap-iron is never dependable, due to a certain amount of foreign scrap that gets into our back-stock. The amount and condition of steel in the charge also affect the ultimate product, and the composition of coke varies over a considerable range. When melting back-stock for mixture with blast-furnace iron, we use about 1 part of coke to 6 parts of metal. The analysis of iron for the cylinders that is found to keep porosity at a minimum and at the same time permit easy machinability is as follows:

Silicon, per cent	2.20
Phosphorus, per cent	0.35
Sulphur, per cent	0.08
Carbon, per cent	3.30
Manganese, per cent	0.80

The process of manufacture as outlined serves us today, but it is continually being developed to a finer degree in accord with the policy of the company. The difficulties due to the human element have been reduced to a minimum and, although there are still some severe working conditions on the cylinder line, they are being eliminated one at a time. The hand-ramming probably will be replaced by a machine operation, upon which we are now experimenting; the cooling of the cylinder will take place while the latter is in motion, the present cooling-trays being eliminated. The experimental molding-machine is shown in Fig. 20. Sand is thrown into the mold at a considerable velocity from the periphery of a revolving wheel, and it seems to pack satisfactorily.

MACHINING THE CYLINDER

The cylinder block, as it comes from the foundry, is inspected for any foundry defects before being removed from the conveyor. The block is not allowed to touch the floor at any time in the actual operation, but moved from one machine to another by conveyors that are as nearly of the same height as the machines as possible. Two of these conveyors are shown in Fig. 21.

The first step in the machining operation is to locate four spots on the upper side of the block. This is done in two specially designed machines. The block then goes to a standard milling-machine to have the crankcase flange milled. After this the six main bearing boltholes, which serve as locating points throughout the entire operation of machining the cylinder block, are drilled and reamed. The top or head side of the block and the water-connection and the manifold side are next milled in one operation. The block is then placed on a four-spindle drilling-machine and a rough-cut taken out of the cylinder bore so that the casting can be tested under a water-pressure of 65 lb. to eliminate any castings that may be porous and will not stand pressure.

Several smaller milling and drilling operations come next. After this the block goes to multiple-spindle drill-

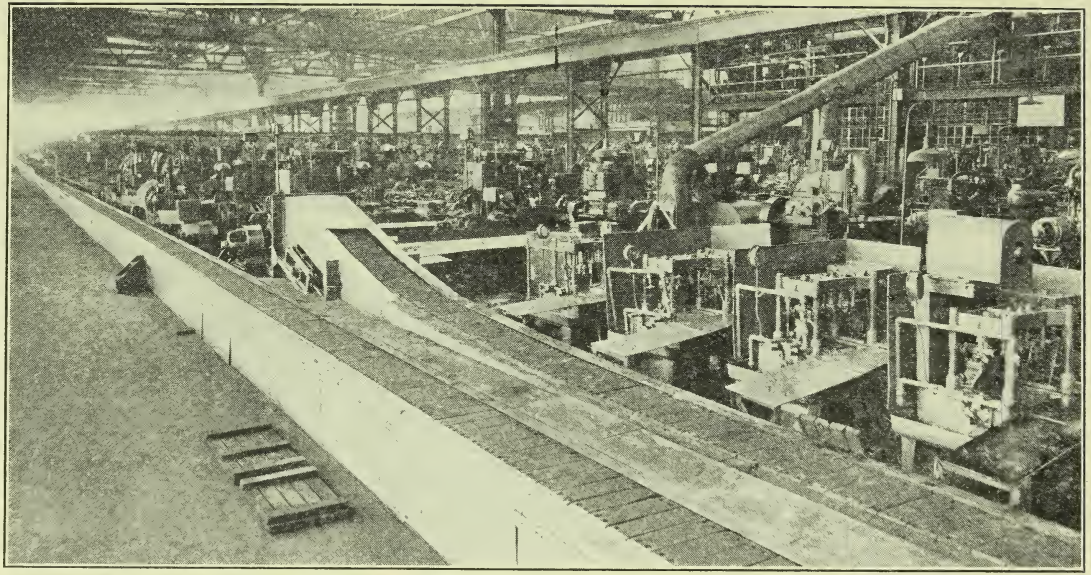

FIG. 21—GENERAL VIEW OF THE CYLINDER MACHINE-SHOP

ing-machines to have the port, valve-stem and push-rod holes bored. All of these machines are equipped with suitable fixtures so that accurately finished work can be turned out rapidly. The radius for the transmission cover is then turned on a standard lathe having special equipment to load and turn two blocks at once. Boring of the camshaft holes is next in order and this is done on a special machine that works from both ends of the block. The block is rebored and also reamed on a four-spindle machine that is similar to the one used to take the first rough-cut out of the bore, some changes in the feed and the speed of the tools of course being made on account of the difference in the nature of the work. The block is then ready for the three-way machine that drills five holes in the top, eight in the manifold side and two in the water-connection side. The block is then put in a four-way machine that drills 15 holes in the top, 4 in the front, 5 in the rear and 17 in the crankcase flange, the entire operation being completed in about 50 sec.

The next major operation is casting the babbitt in the main and crankshaft bearing and milling off the gates in a conveyor machine of special design. The babbitt is then planed to insure the proper fit in the cylinder casting and the correct density of the metal. The next step is the very important one of rolling or glazing the bore, which is done with a four-spindle standard machine driven by a reversible motor. The rolls, which are of special design, are ground very accurately and hardened very uniformly. The block is then placed in a specially designed two-way tapping-machine where 10 holes on one side and 2 on the other are threaded simultaneously. The next and last stage in the machining is the tapping of 15 holes in the top of the block, 4 holes in the transmission end and 3 holes in the front end at the same time by a three-way tapping-machine. From the completely machined blocks we receive approximately 70 tons of chips per day, which are returned to the foundry and immediately remelted to provide the cupola iron needed to make more castings.

After being completely machined the block goes through a specially constructed washing-machine and comes out on a conveyor to be inspected and given the final water-test. If it passes the inspection and test satisfactorily the cylinder block is stamped "ok" and travels along on a conveyor to the loading dock whence it is shipped to the assembling plant.

MALLEABLE-IRON DRILLING DATA[1]

By H A Schwartz[2] and W W Flagle[3]

After commenting upon the two contradictory attitudes toward malleable iron in the automotive industry and outlining its history briefly, the authors discuss the differences between malleable and ordinary gray-iron and supplement this with a description of the heat-treating of malleable castings.

Five factors that influence the machining properties of malleable-iron are stated. These were investigated in tests made with drills having variable characteristics that were governed by six specified general factors. Charts of the results are presented and commented upon in some detail, inclusive of empirical formulas and constants and deductions made therefrom.

The machining properties of malleable-iron is a new subject in engineering literature. C. F. Kettering, past-president of the Society, once expressed the belief that the future of engineering would consist of a careful study of all the materials of construction and the selection of the material for a given purpose the properties of which most nearly correspond to the ideal. The department over which I preside was organized with the idea of furnishing the automotive or any interested industry authentic information as to the properties of malleable-iron castings.

Apparently, there are two well-defined and contradictory attitudes toward malleable-iron in the automotive industry, one being decidedly unfavorable. Having had occasion recently to buy a car, I inquired regarding a certain make of machine. The salesman told me that no malleable castings were used in that car. The first thing observable on raising the hood was a malleable casting and, as a matter of fact, the car in question had 31 parts made of malleable-iron. It is unfortunate that anyone should wish to conceal the use of so valuable a material in parts for which it is suited. The opposite viewpoint is held by certain manufacturers. They seem to feel that malleable-iron should give satisfactory service for any purpose whatsoever, and attempt to make everything of it regardless of its suitability.

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[1] Cleveland Section paper.

[2] M.S.A.E.—Manager of research, National Malleable Castings Co., Cleveland.

[3] Engineer of tests, research laboratory, National Malleable Castings Co., Cleveland.

The history of malleable-iron dates back to Reaumur, who published a description in 1722 of a system of making it by decarburization. In trying to practise the art of making malleable-iron castings, an American named Boyden discovered a different product in 1826 which is now known as black heart malleable. The most recent and accurate figures at my disposal indicate that there are 176 producers of malleable castings in the United States. Others estimate that there are more than 200 producers, but the important ones number between 75 and 80. The principal plants are located in the territory north of the Ohio and east of the Mississippi rivers.

MALLEABLE AND GRAY-IRON DIFFERENCES

Malleable-iron consists of a mass of nearly pure iron or ferrite, through which some 2.0 to 2.5 per cent of carbon is scattered in a spheroidal form in the free state. This particular form of carbon is characteristic of this product only and is known as temper carbon. Gray-iron consists of a metallic mass composed of a mixture of ferrite and pearlite, through which some 3.00 to 3.25 per cent of free carbon is scattered in the form of flaky graphite crystals. It is obvious, as shown by the photomicrographs reproduced in Figs. 1 and 2, that the former conditions produce much less interruption of matrix than the latter. A further consideration is that pure iron is much more ductile than pearlite, which has a corresponding effect upon the two cast products.

The carbon in malleable-iron exists in the geometric form that characterizes it because that carbon is liberated at a temperature when the metal is nearly solid, but the graphite of gray-iron is liberated at a temperature but little below that of the melting point. The fact that it must grow in a nearly solid medium rolls or crushes up the free carbon of malleable-iron into the spheroidal form that characterizes temper carbon. The process of manufacture is first to produce a casting that contains no free carbon and then to heat-treat that casting so as to break up the combined carbon into iron and free carbon.

THE HEAT-TREATING PROCESS

Two fallacies are encountered frequently with respect to the annealing or heat-treating process. The first is

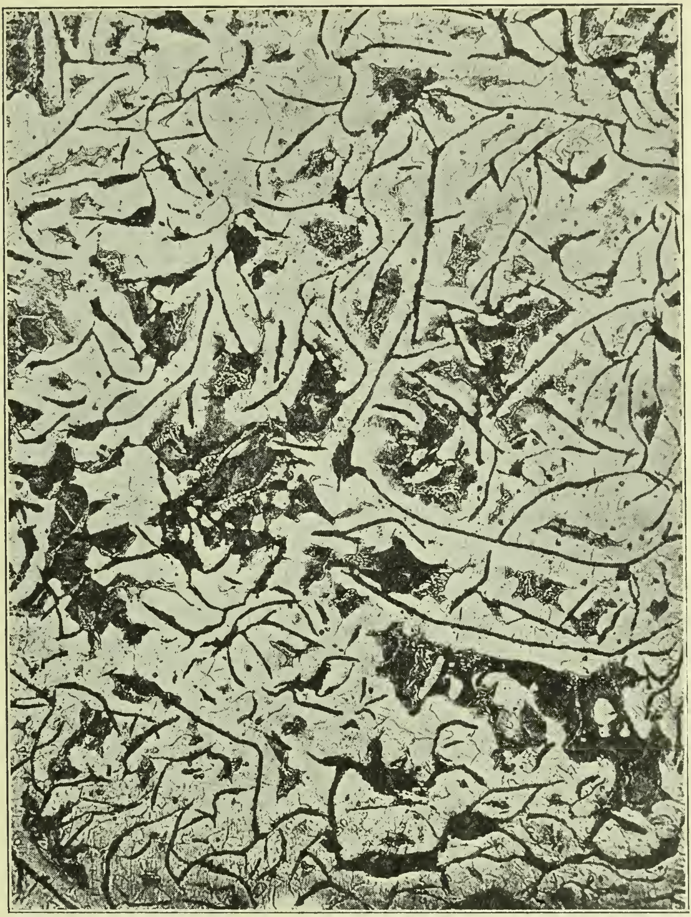

Fig. 1—Photomicrograph of Gray Iron Magnified 100 Times

that the process is conducted for the purpose of eliminating carbon and that, therefore, the surface of the metal must differ widely in properties from those of the center. The elimination of carbon from the surface metal is a mere incident in the process and affects the metal but slightly, increasing the ductility of the product a little. The primary purpose of the heat-treatment is the separation of cementite into ferrite and carbon, and this process does not proceed more rapidly or more completely at the surface than within. Malleable castings, when machined, therefore possess properties that are comparable with those of unmachined castings.

The second fallacy is that the annealing reaction is similar to that used for the annealing of steel and, therefore, the malleable-iron manufacturer is taking too much time for this process. An automotive engineer of my acquaintance once insisted most strongly that we were wrong in taking 9 days to heat-treat castings. He said he would prove this to us by taking a casting in the evening and returning it completely annealed in the morning. That was more than a year ago and he has not yet returned. Steel can be annealed in a few hours, but the various stages in the graphitizing heat-treatment require definite and specific times and cannot be executed in a shorter time interval. All malleable-iron producers would arrange to graphitize the carbon completely overnight if that were possible. The process constantly is subject to study and experiment with a view to decreasing the time involved. So far, however, no great reduction has been found possible.

An attempt to hurry the annealing process results in the user's obtaining an inferior product which impairs the reputation of the producer. Those who purchase material should remember that long annealing processes are executed at the expense of the manufacturer; obviously, they would not be carried out if they were not essential to the satisfactory completion of the product. It is absolutely necessary that demands for malleable-iron products be adequately anticipated to allow sufficient time for this process of manufacture.

The physical properties of normal malleable-iron under various circumstances are covered in my previous papers entitled Malleable Iron as a Material for Engineering Construction[4]; Some Physical Constants of American

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[4] See *Transactions of the American Foundrymen's Association*, vol. 27, p. 373.

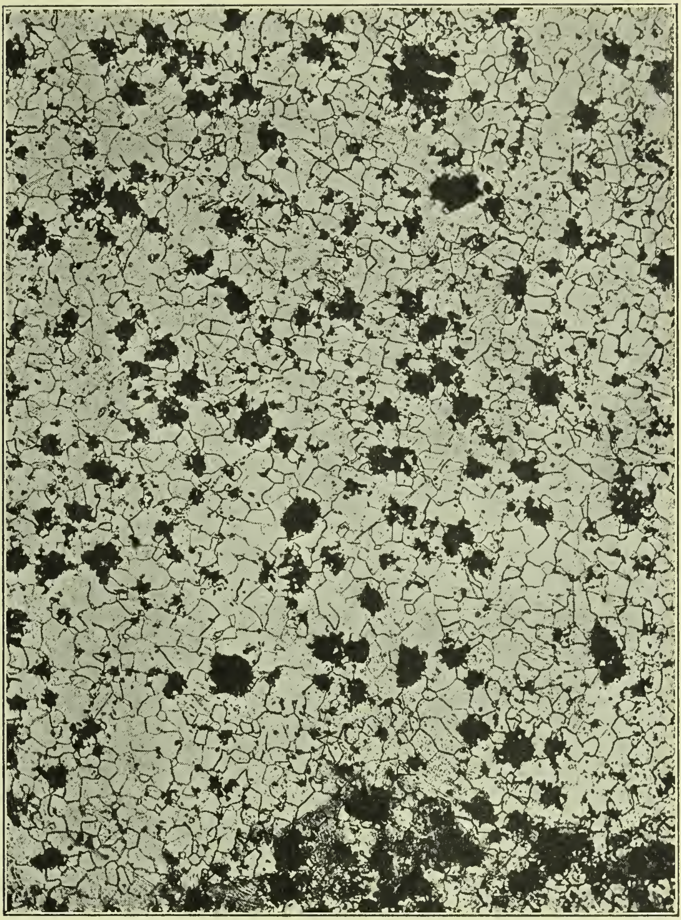

FIG. 2—PHOTOMICROGRAPH OF MALLEABLE CAST IRON MAGNIFIED 100 TIMES

Malleable-Iron[5]; and The Effect of Machining and of Cross-Section on the Tensile Properties of Malleable-Iron.[6]

MACHINING PROPERTIES

The machining properties of malleable-iron are the subject of experiment in the research laboratory of the National Malleable Castings Co. The behavior of twist drills on other products has been the subject of experiment more especially by B. W. Benedict and W. P. Lukens, who gave data obtained in drilling gray-iron in their report entitled An Investigation of Twist Drills.[7] The only data on malleable-iron of which we have knowledge were secured in a very short investigation by Edwin K. Smith and William Barr, and reported in a paper on The Relation Between Machining Qualities of Malleable Castings and Physical Tests.[8]

Feeling that further work was requisite on drilling stresses when cutting malleable-iron, it was decided to begin the study of machinability in general by an investigation of these stresses. The variables, the effects of which are to be studied, include the following, those numbered from 1 to 6 being in reference to the drill and those from 7 to 11 in regard to the properties of the material.

(1) Diameter
(2) Rate of feed
(3) Speed
(4) Point angle
(5) Clearance angle
(6) Helix angle
(7) Chemical composition
(8) Tensile-strength
(9) Elongation
(10) Brinell hardness-number
(11) Shore hardness-number

Since the life of the drill is not under observation, the chemical and physical properties of the drill steel were not significant and were assumed to be constant throughout. The effect of cutting compounds was not within the scope of the investigation.

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[5] See *Proceedings of the American Society for Testing Materials*, vol. 19, part 2, p. 247.

[6] See *Proceedings of the American Society for Testing Materials*, vol. 20, part 2, p. 70.

[7] See University of Illinois, Engineering Experiment Station, Bulletin No. 103.

[8] See *Transactions of the American Foundrymen's Association*, vol. 28, p. 330.

The experimental procedure was to determine on the Olsen universal efficiency machine the torque and thrust of drills operating under various predetermined conditions. Reference is made to a paper by T. Y. Olsen on An Efficiency Testing-Machine for Testing Taps and Dies.[9] All the drills used were of high-speed steel, made and ground by the Cleveland Twist Drill Co. No drill was used to a point where a change in stress that could be detected was produced. This factor was checked by drilling a standard malleable-iron piece from time to time, under standard conditions, as the investigation progressed.

For the investigation of items Nos. 1, 2 and 3, a large amount of malleable-iron from a single heat and a single annealing pot was available. The investigation comprised the drilling of this material with twist drills of standard form having diameters of 7/16, ½, ⅝, ¾, ⅞ and 1 in. at feeds of 0.0025, 0.0050, 0.0100, 0.0200 and 0.0400 in. per revolution at a speed of 240 r.p.m.; and the drilling of the same material with a standard ¾-in. twist-drill at the same range of feeds at speeds that were, as nearly as practicable, 40, 80, 160, 320 and 640 r.p.m. A few runs were made with drills of other diameters to corroborate the conclusion that the effect of speed did not vary greatly with drill diameter.

For the investigation of items Nos. 4, 5 and 6, a new lot of malleable-iron was employed which, from tests made with standard twist-drills, was known to be identical with the first in resistance to drilling. Instead of using the standard drill, which has a point angle of 118 deg., a clearance angle of 12 deg. and a helix angle of 27.5 deg., nine special drills, each ¾ in. in diameter, were provided, that had point angles of 98, 118 and 138 deg. and clearance angles of 5, 10 and 15 deg., respectively. The helix angle was 27.5 deg. in each case. A straight-fluted ¾-in. drill having standard point and clearance angles was provided also.

For the investigation of items Nos. 6 to 11 inclusive, 179 specimens of regular and special malleable-iron were available. These were made by the several plants of the National Malleable Castings and the Eastern Malleable Iron companies and by the Dayton, the Northern, the Erie and the Trenton Malleable Iron companies. Our thanks are due these organizations for placing at our disposal material of divers origins and processes of manufacture. This material represented practically the complete range of quality commercially attainable in the product. Some low-grade specially made material was also used. The metal was all completely graphitized. The chemical composition and tensile properties were determined by the foundry. The Brinell and Shore hardness-numbers were obtained in our own laboratory.

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[9] See *Proceedings of the American Society for Testing Materials,* vol. 14, part 2, p. 541.

Each material was tested with two standard ½-in. drills at 240 r.p.m. and a 0.005-in. feed. The material was drilled also by each of two ½-in. drills running at 240 r.p.m. with a constant pressure of 220 lb. on the

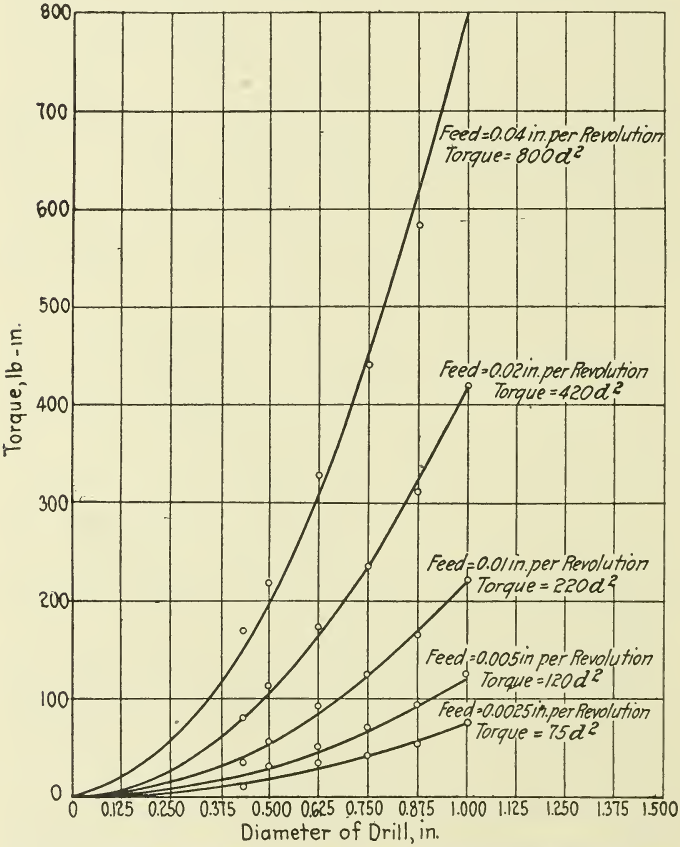

FIG. 3—VARIATION IN THE TORQUE WITH CHANGES IN THE DIAMETER OF THE DRILL AND THE RATE OF FEED

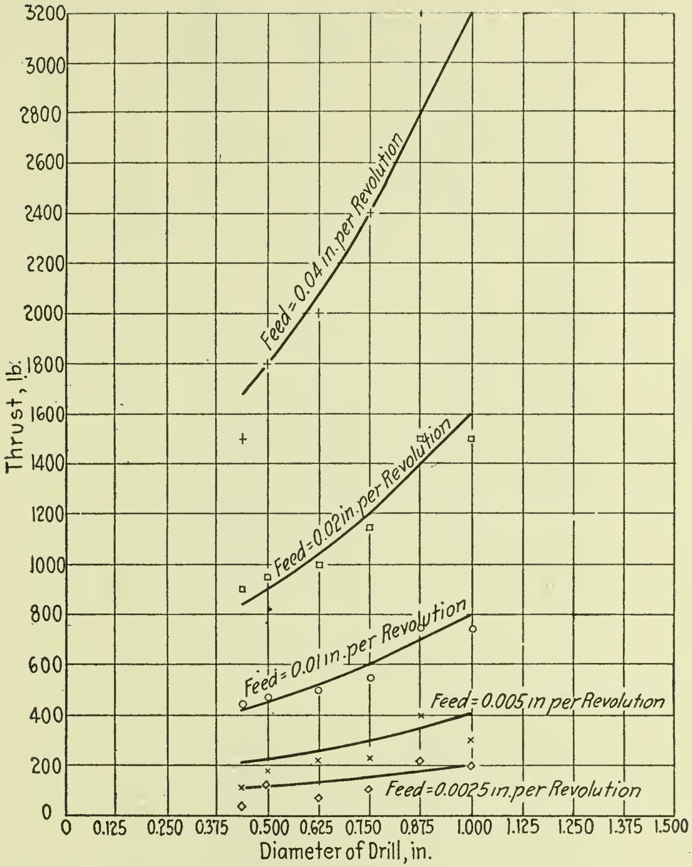

FIG. 4—VARIATION OF THE THRUST, WHICH IS ASSUMED TO BE A CURVILINEAR FUNCTION OF THE DIAMETER OF THE DRILL, WITH CHANGES IN THE RATE OF FEED OF THE DRILL AND ITS DIAMETER

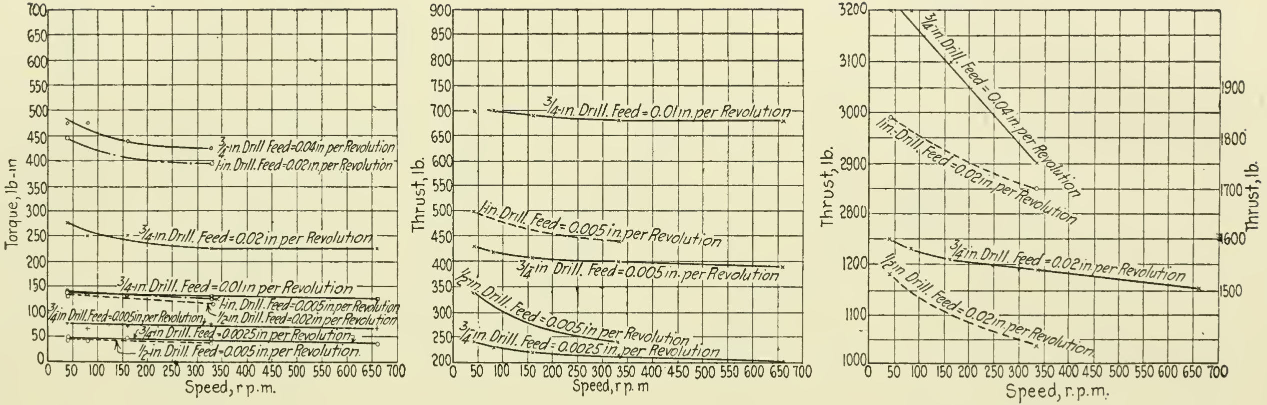

FIG. 5—TORQUE AND THRUST OF A ¾-IN. DRILL AT DIFFERENT SPEEDS AND FEEDS

Results of Some Tests Made with ½ and 1-In. Drills That Were Conducted To Correlate and Corroborate the Conclusions Based on the Tests of the ¾-In. Drill Are Also Presented

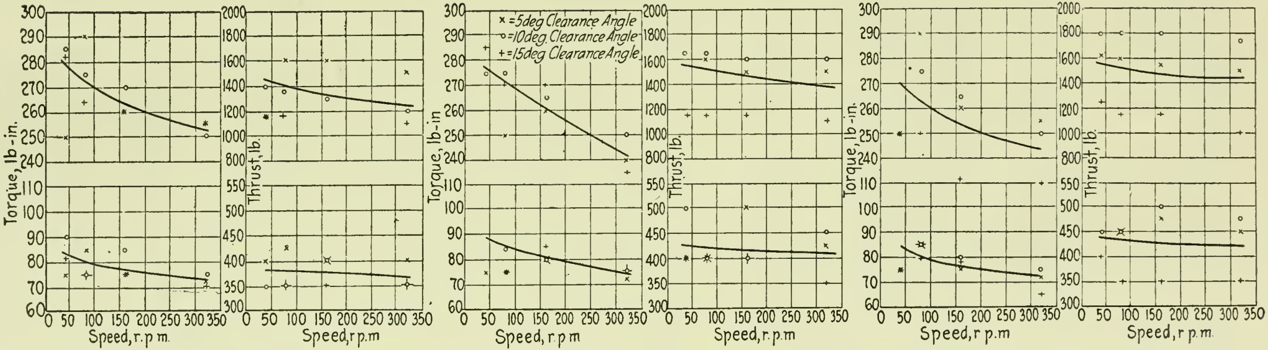

FIG. 6—RELATION BETWEEN THE TORQUE, THRUST AND SPEED OF DRILLS HAVING VARIOUS POINT ANGLES

The Point Angle in the Two Sets of Curves at the Left Was 98 Deg., in the Middle Pair It was 118 Deg., and in the Curves at the Right It Was 138 Deg.

drill point; the torque and the penetration per revolution were recorded. It is obvious that the detailed results of such an investigation are too voluminous to be given and the essential data have therefore been reduced to graphic form.

GRAPHIC TEST-RESULTS

Fig. 3 shows the relation of torque to diameter and feed as a series of parabolas, one for each rate of feed, correlating the diameter and the torque.

Fig. 4 shows the relation of the thrust to the diameter and the feed as a series of flat curves, one for each rate of feed, correlating diameter and thrust. The thrusts developed by the smaller drills, especially at low rates of feed, are so low in value as to render the data somewhat uncertain.

Fig. 5 shows the relation between the torque and the thrust of a ¾-in. drill as related to the speed, a separate line being shown for each rate of feed. A few tests for

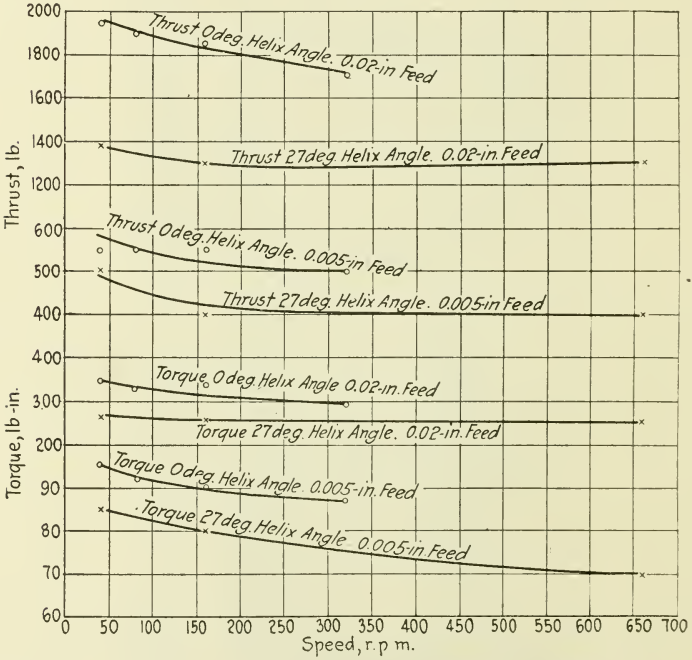

FIG. 7—TORQUE AND THRUST OF STANDARD TWIST AND STRAIGHT FLUTED DRILLS IN MALLEABLE IRON

½ and 1-in. drills were conducted, tending to correlate and corroborate the conclusions based upon the ¾-in. drills.

Fig. 6 shows the torque and the thrust of drills of various point and clearance angles as related to speed. Each chart applies to a drill of a given point-angle, the thrust or the torque being plotted against the speed. A different symbol was used for each of three clearance-angles.

Fig. 7 shows the torque and the thrust of the straight-fluted drill plotted against the speed at two different rates of feed and, for comparative purposes, the same data for a standard twist drill. Thus, Figs. 3 to 7 record the available data on the drilling conditions as a variable and permit fairly accurate conclusions as to the load conditions on drills of known diameter and form, working on the standard material under known conditions of feed and speed.

Effect of the Material Being Machined

In considering the effect of the material being machined as comprised in items Nos. 7 to 11 of the program of investigation, it must be borne in mind that these five factors are not entirely independent variables. Assuming complete graphitization, which is justifiable with good commercial metal and known to apply to our present material, the physical properties of the product are determined primarily by its carbon-content. Were it not for the minor effects of variations in other chemical elements present and in thermal history, the last four variables would be functions of the carbon-content alone and would bear definite relations to one another.

The tensile-strength, elongation and Brinell hardness number increase as the carbon decreases regularly, even under commercial conditions. The Shore number is practically constant for all samples investigated. Therefore, it was to be expected that graphs plotting drilling stresses against items Nos. 7 to 10 would be geometrically similar, and the Shore number would find no application in determining the physical properties of the material. Our tests were all made under such conditions that only material at least ⅛ in. from the surface was examined. Under these circumstances, no connection between decarburization and machinability could be traced, even if it existed. Item No. 7 can thus be dismissed with the statement that items Nos. 8 to 10 are expressions of this variable.

Item No. 11 can be discarded on account of the following considerations affecting the Shore number. The usual physical properties of malleable iron are determined primarily by the relative amount of ferrite and temper carbon in the mass. The Shore test, however, is made upon an almost microscopic area comprising only ferrite, and it is therefore a measure of the properties of the ferrite only. These properties would be affected but little by the presence of the usual amounts of alloy.

In Fig. 8 the thrust and the torque of the ½-in. drill running at 240 r.p.m. are plotted against the tensile-strength and Brinell hardness number, each point representing the average drill-stress for a group of specimens of constant strength or hardness as the case may be. A similar compilation against elongation is omitted as superfluous, since elongation and tensile-strength are interdependent and the latter property is measurable more accurately. Omitting the derivation of the several formulas in the interest of brevity, the data of Figs. 3 to 8 lead us to the following general conclusions that apply to completely annealed malleable-iron castings.

Let

a = A constant depending upon the feed
B = The Brinell hardness number of the malleable iron
b = A constant depending upon the diameter of the drill
d = Drill diameter in inches
f = Rate of feed in inches per revolution
P = The thrust in pounds
s = Speed in revolutions per minute
T = The drill torque in pound-inches
U = The ultimate-strength of the malleable iron in pounds per square inch
W = The work done in drilling, in foot-pounds per cubic inch

The values of *a* and *b* are shown in Fig. 9. Then, for drills of standard form, we have

$$
\begin{aligned}
T &= (0.0049\,B + 0.409) \times (167.06/s^{0.04597})\,a \times fd^2 \\
P &= (0.00755\,B + 0.0952) \times (937/s^{0.03450})\,b \times fd \\
W &= 8\,(0.0049\,B + 0.409) \times (167.06/s^{0.04597})\,a + \\
&\quad (0.00755\,B + 0.0952) \times (937/s^{0.03450}) + (4b/\pi d)
\end{aligned}
$$

where *T*, *P* and *W* are in terms of Brinell hardness number. Or, in terms of ultimate-strength, we have

$$
\begin{aligned}
T &= (0.0000135U + 0.297) \times (167.06/s^{0.04597})\,a \times fds^2 \\
P &= (U/52{,}000) \times (937/s^{0.03450})\,b \times fd \\
W &= 8\,(0.0000135U + 0.297) \times (167.06/s^{0.04597})\,a + \\
&\quad [(U/52{,}000) \times (937/s^{0.03450})] + (4b/\pi d)
\end{aligned}
$$

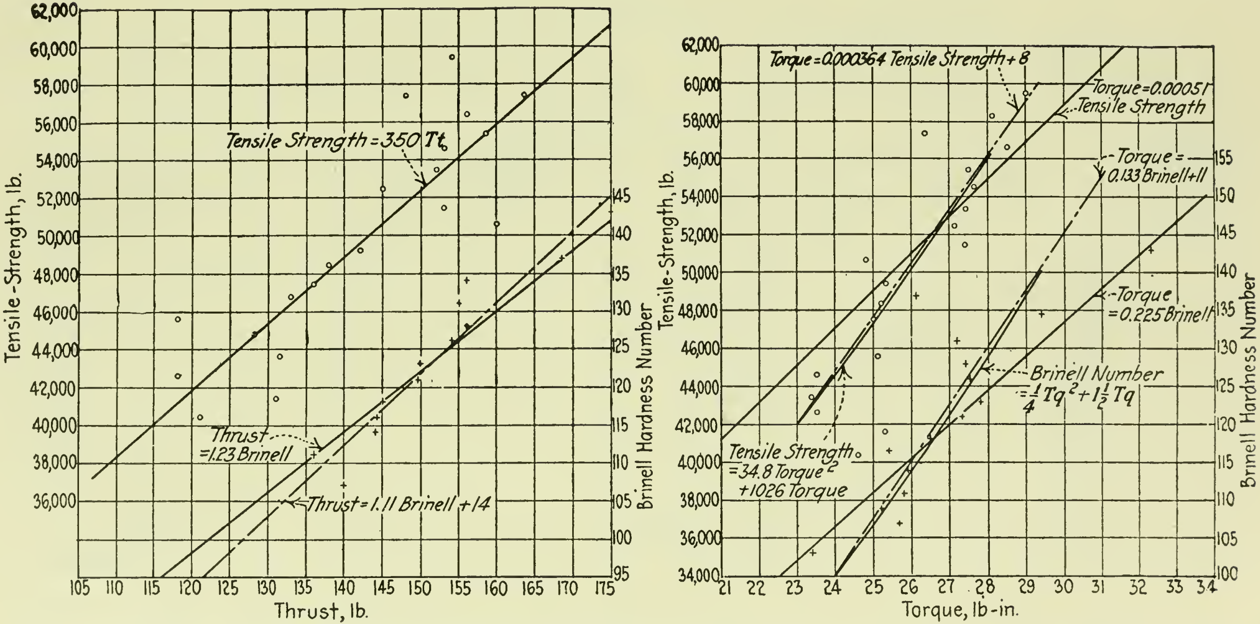

FIG. 8—AT THE LEFT, CURVES SHOWING THE RELATION BETWEEN THE THRUST OF A ½-IN. DRILL AND THE BRINELL HARDNESS-NUMBER AND THE TENSILE-STRENGTH OF THE IRON, AND AT THE RIGHT SIMILAR CURVES FOR THE TORQUE

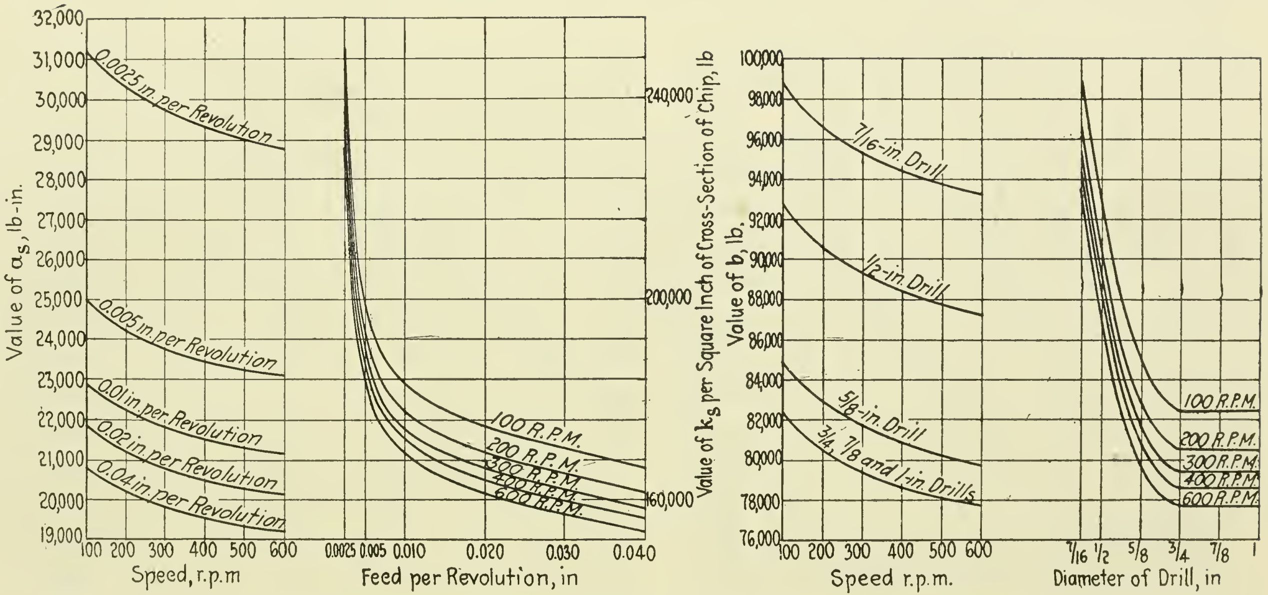

FIG. 9—CURVES GIVING THE VALUES OF a_s, k_s AND b FOR VARIOUS SPEEDS

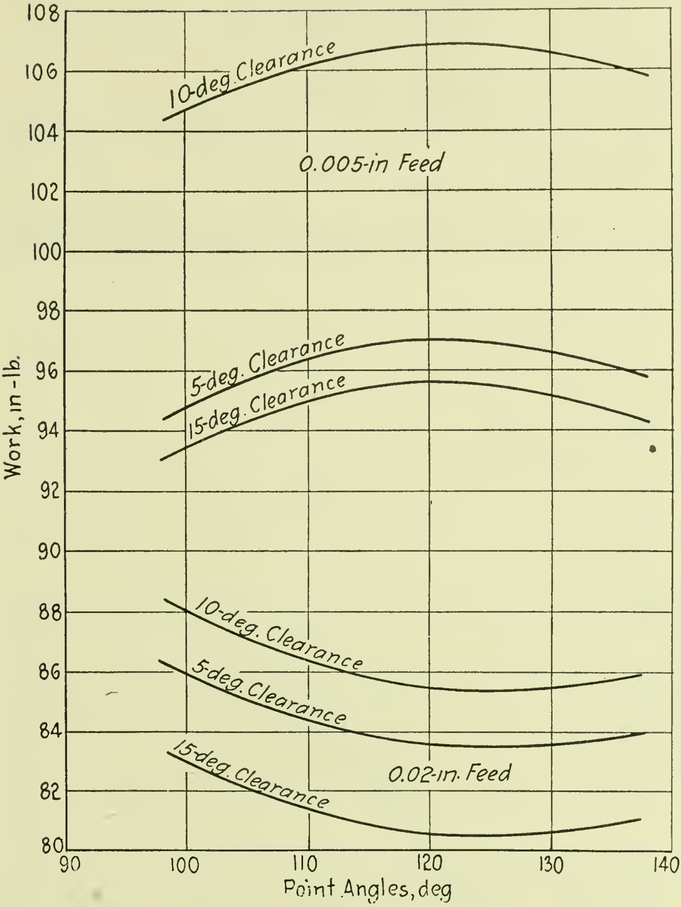

FIG. 10—CURVES GIVING THE AMOUNT OF WORK REQUIRED TO PENETRATE MALLEABLE IRON A DISTANCE OF 1 IN. WITH ¾-IN. HIGH-SPEED DRILLS OF DIFFERENT POINT AND CLEARANCE ANGLES

To simplify these equations, let

$$(167.06/s^{0.04597})\ a = a_s$$
$$(937/s^{0.03450})\ b = b_s$$
$$0.0049B + 0.409 = t_h$$
$$0.00755B + 0.0952 = p_h$$
$$0.0000135U + 0.297 = t_u$$
$$U/52{,}000 = p_u$$

Substituting these values, we have

In Terms of Hardness	In Terms of Ultimate-Strength
$T = t_h \cdot a_s \cdot fd^2$	$= t_u \cdot a_s \cdot fd^2$
$P = p_h \cdot b_s \cdot fd$	$= p_u \cdot b_s \cdot fd^2$
$W = 8t_h\, a_s + 4\ (p_h \cdot b_s/\pi d)$	$= 8t_u\, a_s + 4\ (p_u b_s/\pi d)$

For convenience, values of a_s are plotted at the left of Fig. 9; of b_s at the right of the same illustration; of t_h

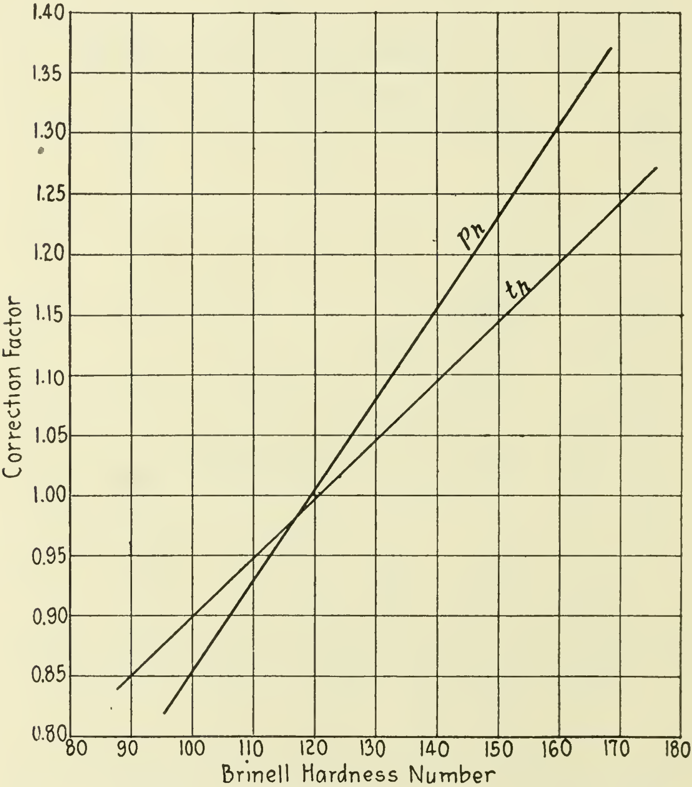

FIG. 11—CORRECTION FACTORS FOR p_h AND t_h FOR VARIOUS BRINELL HARDNESS-NUMBERS

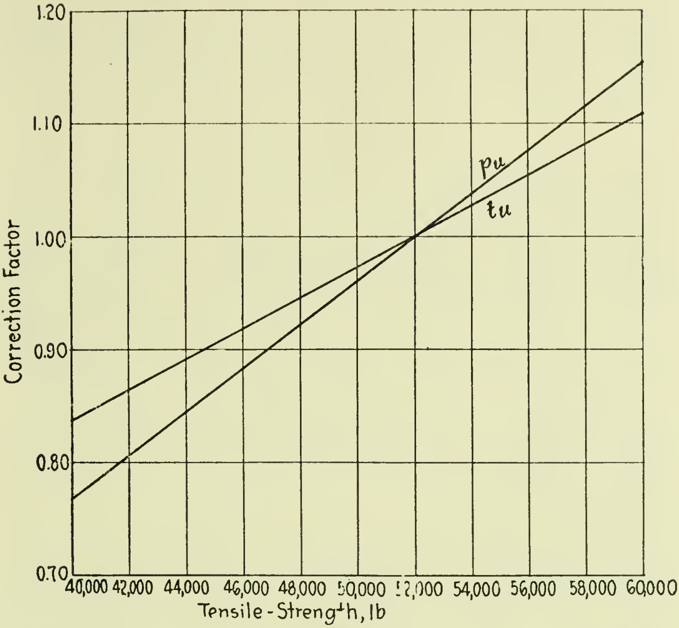

FIG. 12—CORRECTION FACTORS FOR p_u AND t_u FOR DIFFERENT TENSILE-STRENGTHS

and p_h in Fig. 10; and of t_u and p_u in Fig. 11. From these values, T, P and W can be calculated for known values of the several constants.

That the torque should be proportional to the feed and the square of the diameter; that thrust is proportional to feed and diameter; and the derivation of W from T and P, are based on considerations of applied mechanics. The numerical constants are, of course, purely empirical.

In Fig. 12 the relation between work and form of drill point is summarized, as calculated from Fig. 6. Generalizations as to drill form are as yet hardly warranted. From considerations of mechanical efficiency, an increase in the helix and the clearance angles to the highest values consistent with drill wear and strength would seem to be indicated. The effect of the point angle seems unexpectedly small.

By substituting in the formulas for t_h, p_h, t_u and p_u, the extreme values of B and U likely to be encountered it will be observed that the difference in torque between the weakest and strongest irons reasonably likely to be obtained is approximately 30 per cent of the average value. The thrust may vary perhaps 40 per cent between the strongest and weakest commercial irons.

THE DISCUSSION

W. R. STRICKLAND:—What progress has been made in casting thin sections?

H. A. SCHWARTZ:—Malleable iron originally was developed for use in small, thin castings. The recent tendency has been toward heavier sections. I have seen castings where perhaps 20 would be required to weigh 1 oz.; however, such work is very unusual. To avoid cooling strains that may crack the casting, it is necessary to avoid abrupt changes from thin to thick sections.

A MEMBER:—The aluminum foundries have become expert in the use of chills to prevent the cracking of castings. Is there any such art in the making of malleable castings?

MR. SCHWARTZ:—Chills have been used for many years to keep out cracks and also to remove large shrinks. Perhaps I should have spoken of this more fully. One can make perfectly good malleable-iron and still make a very bad casting. For instance, in the automotive trade malleable-iron, when properly produced, is the best possible material for making hubs; but when improperly made, it is about the worst. The reason is that if the work is entrusted to a producer who desires to make the most castings for the least money irrespective of quality, a hub may be produced that is 95 per cent hub and 5 per cent air; the 5 per cent represents the volume of shrink. Being at the point of junction of the barrel of the hub with the flange, these shrinks weaken it at its most critical point and easily may be a cause of failure. The remedy is not in better iron, but in better foundry practice in avoiding shrinkage by the use of suitable feeders. This method, however, costs money and is not likely to be practised by those who feel it necessary on account of competition to cut prices to the limit.

A MEMBER:—Have you experimented with malleable-iron pistons? How much will electric-furnace practice increase the cost of castings? Is the element of manganese considered important?

MR. SCHWARTZ:—Our company has not experimented with pistons. I think malleable-iron possesses no particular advantage. I will not undertake any statement as to the relative cost of electric-furnace and air-furnace malleable. Electric-furnace malleable has been sold continually in competition with the good grades of air-furnace metal. Manganese is of importance on account of the unavoidable presence of sulphur; but I warn any consumer against specifying the chemical properties of the material he desires to buy. The user buys the material on account of its physical or engineering qualities and is interested in the material possessing the qualities that are required. He is usually not well informed as to the method of manufacture, certainly not as well informed as the producer, and should refrain from telling the latter how to secure the results.

A MEMBER:—Is malleable-iron suitable as a bearing metal, for heavy loads?

MR. SCHWARTZ:—Malleable-iron is not suitable for bearings, but there is a specific case in the design of the trucks under a freight or passenger car where the journal-boxes ride against the truck column. In this construction malleable-iron is used. In other words, its resistance to wear is good enough so that its utility for purposes where wear resistance is an incident will not be destroyed. It should not be recommended purely as a bearing material, so far as we know now.

FERDINAND JEHLE:—A piston must be made of good bearing metal, and have walls of such cross-section that they will conduct away some of the heat. The mere fact that it might be possible to make it very thin would not mean that it would be a good piston.

MR. SCHWARTZ:—I do not see, offhand, how one could cast a thinner piston of malleable-iron than of gray-iron, but I think one would have no trouble with it.

E. T. BIRDSALL:—Does it take longer to anneal castings weighing 50 to 60 lb. and from ½ to 1 in. thick than thinner sections? Would it cost more or less to machine malleable than steel castings?

MR. SCHWARTZ:—The heaviest casting of which I have knowledge weighed 896 lb.; it was a transmission housing on a military tractor. A thickness of ½ in. is not considered thick; about 1 in. is considered moderately thick, although much heavier sections have been made. Thickness, as such, has no connection with annealing time, although for other reasons fairly heavy castings take longer to anneal than small ones. I think that, as a general rule, a malleable casting of ordinary size, including machining, will cost less than the corresponding steel casting, the difference being largely in the cost of the machining.

A MEMBER:—You mentioned the wear of malleable axle-housings on railroad-car service. How does the wear compare with that of steel journal-box housings? Is cast-iron used for this purpose?

MR. SCHWARTZ:—Cast iron has been used, although malleable-iron makes the better journal-box. Cast-steel journal-boxes have been made by our company, but they are not common. I know of no figures showing the comparative wear of the several materials. We are now working along these lines.

A MEMBER:—Is there any way of filling porous places in malleable castings?

MR. SCHWARTZ:—They can be filled with compounds such as Smooth-On. They can be filled also by bronze welding or brazing with the acetylene torch. The producer can fill the pores by acetylene welding with white cast-iron and reannealing the castings. In view of the difficulty of producing perfect welds, it is doubtful whether this practice should be resorted to at critical points in a casting. Our company is opposed to it.

A MEMBER: — Does your company request manufacturers to state where they wish the sprue put?

MR. SCHWARTZ:—Not in exactly that form. As a rule, the buyer of a casting is not sufficiently well acquainted with foundry practice to have an opinion of value on this point, but we welcome an expression from him as to where the important points are from which it is necessary to exclude shrinkage.

A MEMBER:—Is it possible to nickel-plate malleable castings?

MR. SCHWARTZ:—That is a very common practice.

DURALUMIN[1]

By R W Daniels[2]

The author gives a short history and general description of duralumin and quotes the Navy specification of its physical properties as drawn by the Naval Aircraft Factory. The manufacture of duralumin is described and commented upon, inclusive of an enumeration of the improvement in physical properties produced at each stage. The physical properties are stated for annealed, heat-treated and hard-rolled duralumin, and some of the possible automotive applications are suggested, inclusive of wormwheels, bearings, gears, connecting-rods, rims and wheel parts and chassis and body trimming.

A report by the research department of the Fifth Avenue Coach Co. on the results of a test it made on duralumin wormwheels is included and the author details the advantages he claims as being attendant upon the use of duralumin.

Duralumin is an aluminum alloy produced after years of systematic search to fill the demand for a metal combining the lightness of aluminum with the strength and toughness associated with ferrous metals. This condition has been met to a remarkable degree and the resulting physical characteristics make duralumin a most desirable material for extensive automotive application. As the commercial manufacture of this metal in this country dates back little more than 2 years, a short history and general description are given to afford a better understanding of the subject, although some information of this character already has been published.

Duralumin was first made in Germany and was developed by A. Wilm and associates during the years 1903 to 1914. The principal and unusual feature of this alloy is that after it has been hot, or hot and cold-worked, it can be strengthened and toughened further from 40 to 50 per cent by heat-treatment. This heat-treatment is somewhat analogous to that of the heat-treating of alloy-steels and consists of quenching from temperatures below its melting point, followed by an aging process. The increased physical properties are not all produced immediately on quenching, but increase during the subsequent aging. In addition to being made in Germany, the manufacture of duralumin was taken up in England by Vickers, Ltd., prior to the late war. During that conflict its use for structural purposes in connection with aviation brought the material before the eyes of the engineering world. Today duralumin is recognized as occupying the same relative position to ordinary sheet or bar aluminum that heat-treated alloy-steel does to ordinary carbon-steel.

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[1] Cleveland Section paper.

[2] A.S.A.E.—Baush Machine Tool Co., Springfield, Mass.

Duralumin is an aluminum alloy containing copper, manganese and magnesium. Its strength and toughness are comparable with those of mild steel, and are obtained with a specific gravity of 2.81 as against 7.80 for steel. The melting-point is approximately 655 deg. cent. (1211 deg. fahr.), the recalescence-point is 520 deg. cent. (968 deg. fahr.), the annealing temperature is approximately 360 deg. cent. (680 deg. fahr.) and the coefficient of expansion is 0.0000225 per degree of temperature centigrade (1.8 deg. fahr.). The chemical composition of the alloy varies within the following limits: copper, 3 to 5 per cent; magnesium, 0.3 to 0.6 per cent; manganese, 0.4 to 1.0 per cent; and the remainder is aluminum plus impurities. Small quantities of other metals are added sometimes for certain specific reasons. For instance, chromium can be added to increase the burnishing qualities of the metal.

The relative modulus of elasticity of duralumin is about one-third that of steel. The Bureau of Standards gives its value as being between 10,000,000 and 11,000,000 lb. per sq. in. Steel is quoted generally as having a modulus of elasticity of 29,000,000 lb. per sq. in. As the physical properties that can be obtained commercially from duralumin have not had much publicity, the following specification, as drawn up by the Naval Aircraft Factory, is of interest:

MATERIAL SPECIFICATION FOR DURALUMIN

Use.—This specification is drawn to cover the requirements of duralumin sheet, rods and wire supplied to the Naval Aircraft Factory

General.—General specifications for the inspection of material, issued by the Navy Department, in effect at date of opening of bids, shall form part of this specification

Material.—This alloy shall show upon analysis the following chemical content:

	Percentage
Copper,	3.5 to 4.40
Magnesium,	0.2 to 0.75
Manganese,	0.4 to 1.00
Aluminum, minimum,	92.0 ..

Specimens for analysis or test shall be taken from the sheet, rod or wire selected as provided by the inspector

Manufacture.—No scrap shall be used other than that produced in the manufacturer's own plant and of same composition as the material specified

Workmanship and Finish.—The sheets must be of uniform quality; they must be sound, smooth, clean, flat and free from buckles, seams, slivers, scratches and other defects

Material in which defects are revealed by manufacturing operations shall be replaced by the manufacturer, notwithstanding the fact that the sheets, rods or wires have previously passed inspection

Physical Properties and Tests.—Duralumin is to be in the heat-treated condition. Its physical properties are to be as follows:

Specific Gravity, 2.80 to 2.85
Yield-Point in Tension, lb. per sq. in., 25,000
Tensile-Strength, lb. per sq. in., 55,000
Modulus of Elasticity, lb. per sq. in., 9,400,000

Selection of Test-Specimens.—At least one specimen for each of the tensile and bend tests shall be taken from a sheet selected to represent each individual melt of the material

The material shall be furnished in the annealed, quenched or "as-rolled" condition, as specified in the order

When material is ordered either in "quenched" or "as-rolled" condition, specimens for the tensile and bend tests shall be tested in the quenched condition. When material is ordered in the annealed condition, specimens for the tensile and bend tests shall be tested in both the physical condition in which the material is received and also in the quenched condition

Specimens for the tensile and bend tests shall be prepared in accordance with the General Specifications for Inspection of Material issued by the Navy Department, except that the form of test-specimens shall be as shown in a sketch to be obtained upon application to the Naval Aircraft Factory

Tensile-Strength.—Tensile test-specimens cut in any

TABLE 1—TENSILE TEST REQUIREMENTS

Physical Condition	Property	Sheets or Strips 0.05 In. Thick or Less Minimum	Sheets or Strips 0.05 In. Thick or Less Maximum	Sheets or Strips Over 0.05 In. Thick Minimum	Sheets or Strips Over 0.05 In. Thick Maximum
Annealed	Ultimate Tensile-Strength, lb. per sq. in.	25,000	38,000	25,000	38,000
Annealed	Elongation in 2 In., per cent	10		10	
Quenched[3]	Ultimate Tensile-Strength, Minimum, lb. per sq. in.	55,000		55,000	
Quenched[3]	Yield-Point, Minimum, lb. per sq. in.	25,000		25,000	
Quenched[3]	Elongation in 2 In., per cent	18		18	

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[3] Quenched specimens shall not be tested within 4 days after completion of heat-treatment. Annealed specimens shall be tested within 12 hr. after treatment.

direction from the sheets must have the properties specified in Table 1

Bend Test.—Specimens cut in any direction from sheets either annealed or quenched must withstand bending cold through an angle of 180 deg. over a diameter equal to four times the thickness of the sheet, without cracking

Dimensions and Tolerances.—The sheets shall be shipped in the lengths and widths called for in the order. The tolerances given in Table 2 will be allowed on the thickness of the sheets

In duralumin forgings where the sections are heavy, it is advisable to lower the minimum tensile-strength requirements to 50,000 lb. per sq. in.; a proportional increase in elongation will be found. Duralumin is unaffected by mercury, is non-magnetic, withstands atmospheric influences and offers a remarkable resistance to sea and fresh waters. It is affected only slightly by numerous chemicals which, in the ordinary way, corrode other metals and alloys so readily; it does not tarnish in the presence of sulphureted hydrogen; and it takes a polish equal to nickel-plating and remains bright without cleaning longer than any plated or silvered article. It

TABLE 2—ALLOWABLE TOLERANCES FOR SHEETS NOT WIDER THAN 18 IN.

Normal Thickness, in.	Tolerances, in.
0.0808 and more	±0.005
0.0808 to 0.0359	±0.003
0.0320 or less	±0.001

is the ideal substitute for aluminum, German silver, brass, copper, nickel-plated and silvered articles, and is the only substitute for steel where lightness combined with the strength of that metal is required. It is the only light metal that can replace steel in forgings, with a two-thirds saving in weight. Heat-treated duralumin forgings approximate mild-steel forgings in strength. Wherever weight is a deciding factor, duralumin is the most satisfactory metal for most shapes made by hot-working or forging. Naturally, duralumin forgings are especially desirable for reciprocating or moving parts where inertia, due to their own weight, forms a large part of the total stress. Duralumin machines and polishes very easily and, as it does not rust or corrode, it can be used in many places where weight is not the prime essential.

The manufacture of duralumin is somewhat analogous to that of steel and, in brief, is as follows:

(1) Manufacture of the alloy from its aluminum base
(2) Casting the ingot
(3) Hot-rolling or cogging in blooms, billets or slabs
(4) Hot or cold-working to final shape
(5) Heat-treating

The ingots are poured at as low a temperature as is practicable; that is, just enough above the melting-point to fill the mold and prevent cold-shuts. The ingots are then either hot-rolled or cogged into slabs, blooms or billets, similar to the manner of working steel. This hot-working is done at a temperature of from 450 to 480 deg. cent. (842 to 896 deg. fahr.), and care must be used not to perform any work on the metal above these temperatures because there is a danger of hot-shortness if the material is rolled or forged at higher temperatures. It is seen readily that such low temperatures cannot be judged by color; therefore, it is necessary to use accurate pyrometers in heating the metal, previously to working. The final rolling or forging can be done hot or cold, according to the character of the work being handled or the nature of the shape it is desired to produce.

The hot or cold-worked metal in its final shape shows greatly improved physical properties over those of the cast ingot, but the full development of its qualities is obtained only by a specific heat-treatment. To obtain this heat-treatment, the metal is heated to a temperature of 500 to 520 deg. cent. (932 to 968 deg. fahr.) for a period of time, depending upon the section of the piece, and immediately quenched. The heating and quenching immediately start to improve the physical qualities of the metal, but the maximum results are obtained only by the subsequent aging. During the aging period, which takes from 1 to 5 days, the alloy markedly increases in tensile-strength, hardness and elongation. Aging is sometimes accelerated by placing the metal in a hot-water bath up to 100 deg. cent. (212 deg. fahr.), or in a hot room. The above heat-treatment develops the remarkable properties possessed by duralumin, and these properties have not been obtained in like degree in any other aluminum alloy.

IMPROVEMENT OF PHYSICAL PROPERTIES

The various stages of manufacture, as related, increase the physical properties of duralumin by distinct steps. The cast ingot shows a tensile-strength of from 28,000 to 32,000 lb. per sq. in., with an elongation in 2 in. of from 1 to 3 per cent. The hot or cold-worked metal shows a tensile-strength of from 40,000 to 50,000 lb. per sq. in., with an elongation of from 6 to 12 per cent. These last figures are variable, depending upon the amount of working in the cold state. Upon subsequent heat-treatment and aging, the physical properties of duralumin show a marked increase, namely, 55,000 to 65,000 lb. per sq. in. tensile-strength and an elongation of from 18 to 25 per cent.

When it is required to put a considerable amount of work upon duralumin in its finished state, it often is found necessary to anneal the sheets between operations in precisely the same manner as in handling other metals. This annealing should be done at 350 deg. cent. (662 deg. fahr.). If several drawing operations are to be performed, it may be necessary to anneal the metal between such operations. Annealed duralumin can be heat-treated and the maximum physical properties obtained, no matter what the shape or form to which the metal may be reduced. Conversely, heat-treated duralumin can be annealed.

Duralumin can be cold-worked after heat-treatment and aging. This operation produces a hard, smooth finish and materially increases the tensile-strength of the metal at the expense of elongation; that is, the tensile-strength will increase from 6000 to 10,000 lb. per sq. in. over that of the heat-treated metal, but the elongation may drop as low as 3 or 4 per cent.

In the annealed form it can be drawn, spun, stamped or formed into a great variety of shapes, as is the case of brass and mild steel. The physical properties in this state average as follows:

Ultimate Tensile-Strength, lb. per sq. in.,	25,000 to 35,000
Elongation in 2 In., per cent,	10 to 14
Brinell Hardness,	54 to 60
Scleroscope Hardness,	9 to 12

Duralumin in its heat-treated form can be slightly shaped or formed and can be bent cold to 180 deg. over a mandrel four times the thickness of the sheet. Its remarkable tensile-strength is here combined with its maximum elongation as follows:

Ultimate Tensile-Strength, lb. per sq. in.,	55,000 to 62,000
Yield-Point, lb. per sq. in.,	30,000 to 36,000
Elongation in 2 In., per cent,	18 to 25
Brinell Hardness,	93 to 100
Scleroscope Hardness,	23 to 27

Heat-treated duralumin forgings have similar physical properties. Heat-treated and hard-rolled duralumin is used where no bending or forming is required. It is a very hard, strong, springy metal in this state and machines or polishes beautifully. Its physical properties in this form average as follows:

Ultimate Tensile-Strength, lb. per sq. in.,	67,000 to 72,000
Yield-Point, lb. per sq. in.,	55,000 to 65,000
Elongation in 2 In., per cent,	3 to 8
Brinell Hardness,	130 to 140
Scleroscope Hardness,	37 to 42

Having covered the general characteristics of the metal, a more intimate discussion of a few of the many automotive applications is given. Some of these applications are still under experimental observation and, in others, duralumin has been adopted as a standard material.

Automotive Wormwheels and Bearings

During the past 2 years much experimental work has been done along this line and the data are now available. Since the characteristics of the metal brought out in this class of service are highly desirable in other forms of gearing, bushings and the general replacement of bronze, these data are given at some length.

From the general description of duralumin it will be seen readily that here is an ideal material for wormwheels, provided the bearing or wearing qualities are satisfactory. For a given section, the weight is one-third that of the conventional bronze. The tensile-strength and the relatively high elastic-limit assure superior tooth-strength. The homogeneous structure and uniform hardness of heat-treated duralumin forgings obviate hard spots, porosity and spongy areas so common in bronze castings, entailing not only machining losses but uneven tooth-wear in service. The excellent machining qualities assure the manufacturer a saving in the machining costs, compared with those of bronze.

The wearing qualities of wormwheels for automotive purposes are best determined by actual road service, as bench or laboratory-test results do not always correspond. It is instructive, however, to compare results obtained from duralumin with those from other materials under identical conditions. The data from various laboratory tests under my observation on bronze and duralumin wormwheels can be summarized by saying that tests destructive to duralumin wormwheels were also destructive to those made of bronze. The results are always good when duralumin and hardened steel are run together. An example of this application is shown by duralumin connecting-rods running direct on the wrist-pins with better life at this point than do conventional bronze-bushed connecting-rods of equal bearing area.

Comparative tests of bearings made from duralumin against bearings made of genuine babbitt metal show that, for shaft speeds exceeding 700 r.p.m. and loads over 200 lb. per sq. in., duralumin bearings develop less friction, remain cooler and show practically no loss in weight under the most severe conditions. For lower bearing pressures and slower speeds, babbitt metal is superior. Table 3 shows the details of this test.

TABLE 3—COMPARATIVE BEARING TEST

Loads, Lb. per Sq. In.	Speed, R.P.M.	Total Number of Revolutions	Final Temperature		Rise in Temperature		Friction, Lb.	Loss of Weight, Grams
			Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.		
Baush Duralumin, Grade B								
100	632	37,920	39	102	18	64	21.15	.....
200	625	37,500	71	160	70	158	42.30	(*a*)
300	629	37,740	54	129	32	90	63.45	.....
400	623	37,380	62	144	39	102	84.60	(*b*)
Genuine Babbitt, Bureau of Standards								
100	694	12,230	89	192	53	95	22.00	0.023
200	706	16,510	102	216	58	104	29.00	0.021
300	686	15,150	125	257	100	180	38.00	0.013(*c*)
400	603	5,500	139	282	94	169	79.00	0.054(*d*)

(*a*) Bearing roughed and ran warm in 10 min.
(*b*) No measurable loss of weight
(*c*) Belt slipping
(*d*) Bearing seized, smoking

In regard to road tests, a considerable number of duralumin wormwheels are now actually in regular service in trucks ranging from 1 to 3½-ton capacity. These wheels have been in service from a few weeks to over 2 years without any failure. As these wheels are all running, complete data are not available, but through the courtesy of G. A. Green, vice-president and general manager of the Fifth Avenue Coach Co., New York City, I quote from the report of one of its preliminary tests under date of Aug. 2, 1921, as follows:

FIFTH AVENUE COACH CO. REPORT

General.—The greatest possibility of effecting weight-saving lies in the employment of aluminum or some of its alloys. With this idea in mind it was decided to test in road service a rear-axle wormwheel fabricated from an aluminum alloy commercially known as duralumin

Object.—To determine by road test the merits of a duralumin wormwheel, especially noting its resistance to wear, relative weights and the like, as compared with the standard bronze unit

Description.—Duralumin is a light aluminum-alloy having a specific gravity of 2.82. It can be forged, stamped, drawn or spun. The product is highly resistant to corrosion. The metal is heat-treated by the producer in a manner that is not made public. The following physical properties are claimed for this material:

Tensile-Strength, lb. per sq. in.,	55,000
Elastic-Limit, lb. per sq. in.,	30,000
Elongation in 2 in., per cent,	18

Bend cold over 180 deg. on a mandrel four times the thickness of the sheet.

A wormwheel was cut having standard pitch and ratio. The relative weights are, for duralumin, 15.0 lb.; and for standard bronze, 41.5 lb. The difference, 26.5 lb., is the equivalent of 64 per cent

Method.—Three duralumin wormwheels were procured from the Baush Machine Tool Co., and installed in standard worm-carriers. The road test was started on three 2A-type buses. An inspection of these parts was made periodically for the first few weeks of service and again during the next annual overhaul of these buses

Results.—The results obtained with these sample wormwheels are recorded in Table 4.

TABLE 4—RESULTS OF TESTS MADE BY THE FIFTH AVENUE COACH CO. ON DURALUMIN WORMWHEELS

Bus Number	30	39	40
Date Installed	Aug. 27, 1920	Sept. 15, 1920	Sept. 11, 1920
Date Removed	June 20, 1921	In service	June 17, 1921
Mileage	26,672	24,143	32,253
Fuel Average, miles per gal.,	6.75	6.52	6.65

> From the above tabulation it will be noted that a total distance of 83,068 miles has been covered with these units, all of which show excellent resistance to wear along the pitch-line at the end of this period. In one case the unit removed from bus No. 40 had a bearing failure behind the worm. The sides of this worm were slightly chipped, but not sufficiently to prevent returning it to service with the others
>
> *Conclusion.*—Inspection of these parts after the above stated amount of service indicates wearing qualities equal to those of the standard bronze wormwheel. In view of the advantages to be obtained from the use of this material it is recommended that several more be obtained for a more exhaustive test

After long tests with bronze wheels where the oil has not been changed, the oil is found to contain particles of bronze in suspension. This condition is very marked in some tests, and is of importance not alone as indicating tooth wear but as showing the deterioration of the lubricating value of the oil. Oil heavily charged with metallic particles acts more like an abrasive than a lubricant and is an important factor in automotive worm-gear wear, because the oil is seldom renewed as often as is desirable. When duralumin wheels were used, the charging of the oil with metallic particles was practically negligible. As brought out by these tests, indications point to excellent life as well as lightness for duralumin wormwheels, unless the wheels have been roughened by lack of lubrication or too high a tooth-pressure, which will injure or destroy any worm gearing.

GEARS

The same qualities that make duralumin a desirable material for automotive wormwheels make it valuable for plain spur and other gearing. It is suitable for this class of work where the pressures are sufficiently within its elastic-limit of 30,000 lb. per sq. in. Where this condition is met and light weight and quietness are desirable, it replaces iron, steel, brass, fiber, fabric and the like. Where duralumin can be run against steel rather than against itself, the best results are obtained. The outstanding automotive application is found in the timing-gear trains of automobile engines where both long life and quietness are essential.

Helical-cut spur-gears of duralumin, alternated with steel gears, have been most successful in service. Detailed test reports are not especially interesting as the gear design varies with every engine, but the fact that upward of 80,000 duralumin camshaft and idler gears are now in use is conclusive. It may seem somewhat paradoxical that duralumin gears, when meshed with steel gears, are quiet, because all duralumin forgings are resonant when struck. The explanation is undoubtedly in the difference in pitch of the sound vibrations of steel and of duralumin. This, of course, applies only when the mass and section of the duralumin gear are properly proportioned to the steel gear.

Connecting-Rods

Reciprocating and other high-speed parts naturally offer a field for duralumin forgings or shapes, and their performance under alternating stress has proved highly satisfactory. Duralumin connecting-rods give remarkable results in high-speed engines, especially in connection with aluminum or other light-weight pistons. Much experimental work has been and is being done, the data being beyond the scope of this paper; the work really comes under the heading of the effect of light reciprocating parts on engine design. However, it is safe to state that duralumin connecting-rods can replace steel connecting-rods, while retaining the same outside-diameter dimensions.

It is recommended that the radii and sections be somewhat increased, depending upon the characteristics of the steel replaced. However, such a large part of the stress on a connecting-rod is due to its own weight and that of the piston that a considerable sacrifice of tensile-strength is allowable to the duralumin connecting-rod due to the weight saved on the rod. The average I-beam-section steel connecting-rod generally weighs about twice as much as the duralumin connecting-rod that can replace it. The piston-pin can be floated directly in the duralumin connecting-rod but, except in special cases, it probably will be desirable to babbitt the lower end.

OTHER PARTS

One of the most satisfactory applications of duralumin to the automobile is that of the rim and other wheel parts. Here the engineer is not only appreciably reducing the unsprung weight and cutting down the centrifugal action of the wheel, but is giving the owner-driver something he can see and appreciate. First, the minimum saving of 10 lb. for the rim alone is welcome in tire-changing. Second, the non-rusting or non-corroding characteristic of duralumin allows the rim always to function properly and prevents the tire from rusting and vulcanizing to the rim. These advantages, added to the strictly engineering ones, justify the necessary increase in cost of the change from steel in certain grades of car at least. As the physical properties of duralumin actually exceed those of the 0.10 to 0.15-per cent carbon-steel used in rims, cost appears to be the only drawback. A certain type of disc wheel, with discs and rims of duralumin of the same sections as the steel design, have been subjected to a road test for more than a year with entirely satisfactory results, and much experimental work is now going on.

Duralumin is an ideal material for chassis and body trim that requires a bright finish. Great strength and saving of weight are not of prime importance here. A commercially workable material of pleasing appearance, of high resistance to atmospheric influence or other tarnish and not plated, is wanted. Duralumin forgings, stampings and drawings, especially when supplemented by aluminum-alloy castings of the same color and resistance to tarnish, are being used with satisfaction to replace nickel-plate, brass and steel in such articles as hub-caps, door-handles and instrument-board fittings.

It is obviously impossible in a paper of this character and length to do more than describe the physical and metallurgical properties of duralumin and touch on some of the applications of interest to the Society. However, the automobile engineer should realize that now he can avail himself commercially of this extraordinarily light yet strong material as the aeronautical engineer has done. The development of the Zeppelin was dependent upon the development of duralumin, and the rapid progress of the all-metal airplanes since the war has been due largely to the commercial availability of the same metal. While the automobile is not as dependent as the airship or airplane upon a material of this class, nevertheless modern automobile design tends toward the elimination of unnecessary weight. The extensive introduction of duralumin into automobile construction, especially in unsprung and reciprocating parts, will permit of a complete redesign, effecting economies that will offset its greater cost as compared with steel. Such a car will bring nearer the ideal combination of road performance and economy of upkeep and operation.

THE GROUP-BONUS WAGE-INCENTIVE PLAN[1]

By E Karl Wennerlund[2]

The author states that the purpose of every plan of wage incentive is to stimulate the worker to a greater effort than is generally obtained on a straight day's-work basis; to reward him somewhat in proportion to his effort; and to gain other advantages such as greater attention to conditions that curtail production, more uniform labor costs and the elimination of inefficient employes. He states further that nearly all industries engaged in repetitive work are now on an incentive basis.

After outlining the most successful wage-incentive plans and enumerating some of the conditions that must be met, inclusive of four specific fundamental principles of industry that are stated, the group-bonus plan is explained and the application of group standard-time is discussed at some length, supplemented by tabular data. Experience with grouping is then related and conditions favorable to grouping are mentioned. The summary states that the group plan, like any other wage-incentive plan, has for its main objective intensive production.

The subject of wage incentive is an important one to factory executives. During the last few years it has received considerable study. Nearly all factories now have their own specialists who make time-studies of operations and recommend or set production rates. Even when outside industrial engineers are employed for this purpose their services are regarded as temporary, the idea being to develop a local staff as rapidly as possible to handle the regular routine. The question of the particular form of wage incentive, therefore, becomes important. No single plan can be recommended for all factories. Local conditions and the character of product will have to govern in each case. It is, of course, desirable to have an incentive plan handled as simply as possible with a minimum of clerical effort, and with the least liability to error in quantity checking.

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[1] Detroit Production Meeting paper.

[2] Production department, General Motors Corporation, Detroit.

The purposes of every plan of wage incentive are the same:

(1) To stimulate the worker to a greater effort than is generally obtained on a straight day's-work basis

(2) To reward him somewhat in proportion to his effort

(3) To gain other advantages such as the attention given to conditions that curtail production, more uniform labor costs and the elimination of inefficient employes.

The result has been that nearly all industries engaged in repetitive work, with very few exceptions, are on an incentive basis. Until recently, the plan followed has been generally of the type known as the individual-effort plan, with either straight piece-rates or some form of premium or bonus based on time measurements. Grouping of employes was not often resorted to, except for such operations as made it impracticable to keep track of the output from individuals. However, the principles of grouping, with a division of the earnings among several workers, have been well known and have been applied for a long time to such work as steam hammering, riveting, and handling materials; but, for work done distinctly by an individual, the usual practice has been to set a rate for that particular detail operation and to pay the individual worker for his production at that operation. The application of grouping of employes throughout an entire factory organization, wherever work is measurable, is rather a new development in industry.

In automobile factories and most accessory plants, the character of product is such as to make parts move progressively from one operation to the next to a very great extent. Processes have been standardized and divided into minute detail and, being of such repetitive nature, operations have been segregated into progressive flow-lines wherever practicable. This was a natural result from the desire to minimize the expense and delays incident to transporting materials in process. The consequent physical arrangement of these factories has had an important bearing on wage-incentive problems.

It must be taken into account that an individual-effort system of payment to workers entails considerable routine expense. The earliest method, after rates had been set, was for each worker to make out a report at the end of the day stating how many pieces he had completed under each operation; or a shop checker came for this information and inserted the quantities on a prepared form. A systematic check on quantities after each operation was, of course, impracticable with large-volume production unless some special means were provided; so, there was much opportunity for collusion between shop checkers and workers and there is no doubt that many took advantage, particularly on parallel operations, of the management's inability to obtain an actual check. One safeguard often resorted to was to discourage the worker from turning in more than a certain maximum rate of earnings; but, even then, it was no guarantee that amounts up to the maximum might not be fraudulently claimed. It was just about as logical to continue to operate a factory under these conditions as it would be to eliminate the time-clocks at the entrance gate or to do away with cash-registers. To do so would require a lot of faith in human nature. There is no doubt that the time-clock and cash-register have both helped many men to remain honest.

Well-organized factories, operating under the individual-effort plan in recent years, have developed various methods for obtaining physical checks on quantities after each operation. One of these is to have a set of coupons, numbered serially, attached to each large piece or to a container of a pre-determined number of pieces. The worker is then required to present these coupons as evidence of quantities. By another method known as the lot system, the worker obtains a job ticket for each lot. The start and finish time of the operation for each lot is stamped on the job ticket which bears the same serial number as the lot quantity. Later, when these tickets come to the office, each is checked off on a master lot-card, then extended and credited to the individual account.

These methods are all good in theory. No doubt they pay their way when compared with having no definite check, and they may be entirely practicable in small shops or for certain classes of work, but let us consider the modern automobile factory. The lot method not only ties up a large inventory of parts in process, but to check off and extend thousands of job tickets, daily, also involves a great expense. One such factory had in excess of 25,000 job tickets for each working day, each one of which had to be carefully audited as to quantity and rate, extended for total and then credited to the account of the employe.

It was the desire to simplify the factory routine, and to escape from this mass of detail management without sacrificing the principles of quantity checks, that prompted the development of the group-bonus plan. Certain fundamental principles of industry had to be recognized and taken into account. The most important of these are that

(1) Under any incentive plan, the worker must continue to maintain an individual interest in the rate of production
(2) He should be guaranteed a satisfactory hourly wage
(3) He should be paid for all time saved instead of for only a part of it
(4) He should be able to compute his own earnings, accurately and quickly

Under the group plan, it would not make much difference in theory whether a fixed group-price were established or some form of bonus or premium payment were used. Our past experience, however, indicated that it was preferable to set standards of output in *time* rather than in money value. With changing labor scales, we should then have no disturbance of the existing time-standards, but would make the adjustments on the hourly-base wage-rates on which bonus earnings are computed. Time standards also have the advantage of a very flexible wage-rate, by which means new employes can enter our service at a minimum rate subject to advancement through length of service and personal ability. Such revision in wage rates can be made at any time without in the least disturbing shop standards or affecting other employes.

On the other hand, a fixed group-price in money value would offer some very definite objections if adopted as a general plan, since the

(1) Division of the earnings among the group members would have to be on an even basis per hour worked; or

(2) Group earnings would have to be pro-rated on the basis of the hours worked by the individuals, multiplied by the predetermined base-rates. A change in an individual base-rate would affect all other members of the group, or the transfer of an employe with a different base-rate would upset their relative earnings

In regard to a uniform base-rate for all members of a group and their division of the earnings purely on an hourly basis, it is obvious that this method would prove unsatisfactory as a general policy; because, even in the same group, some operations may require more skill than others. Then there is the added consideration that employes who have been long in the service should be on a higher wage plane than beginners. The standard-time bonus-table as adopted offers every facility for differentiating between base-rates and permits a flexible system to suit the varying labor and factory conditions.

Table 1 is based on the theory that a bonus incentive of 20 per cent should be paid to a worker, in addition to his hourly base-rate, for attaining an efficiency of production of 100 per cent; and that, for a greater speed of production, he should be paid for all time saved. The table is constructed so as to give a gradually disappearing percentage of bonus, as efficiency of production falls below the standard. During the last 11 years, this bonus table has been used extensively in our plants; first as an individual-effort incentive and, in later years, as a group incentive. A graphic chart showing the relation between efficiency and percentage of bonus is reproduced in Fig. 1.

The standard time of a single operation is determined by time-study. For this purpose the decimal stop-watch is used. By standard time is meant the time required by the average competent workman, taken repeatedly, over an extended period, eliminating lost and waste time. It will contain allowances for legitimate tool-changes and for rest and delay to the worker. It is our practice, for convenience, to designate standard time in decimal hours rather than in minutes. For this purpose a conversion table has been devised and is presented as Table 2, because the detail time-study is observed in decimal minutes, while the standard time is converted into hours.

Group standard-time is obtained by totaling the individual standard-times for all operation embraced by a

TABLE 1—BONUS TABLE FOR PRODUCTIVE WORKERS

Percentage		Percentage		Percentage	
Efficiency	Bonus	Efficiency	Bonus	Efficiency	Bonus
75	1.0	117	40.4	159	90.8
76	1.6	118	41.6	160	92.0
77	2.2	119	42.8	161	93.2
78	2.8	120	44.0	162	94.4
79	3.4	121	45.2	163	95.6
80	4.0	122	46.4	164	96.8
81	4.6	123	47.6	165	98.0
82	5.2	124	48.8	166	99.2
83	5.8	125	50.0	167	100.4
84	6.4	126	51.2	168	101.6
85	7.0	127	52.4	169	102.8
86	7.6	128	53.6	170	104.0
87	8.2	129	54.8	171	105.2
88	8.8	130	56.0	172	106.4
89	9.4	131	57.2	173	107.6
90	10.0	132	58.4	174	108.8
91	11.0	133	59.6	175	110.0
92	12.0	134	60.8	176	111.2
93	13.0	135	62.0	177	112.4
94	14.0	136	63.2	178	113.6
95	15.0	137	64.4	179	114.8
96	16.0	138	65.6	180	116.0
97	17.0	139	66.8	181	117.2
98	18.0	140	68.0	182	118.4
99	19.0	141	69.2	183	119.6
100	20.0	142	70.4	184	120.8
101	21.2	143	71.6	185	122.0
102	22.4	144	72.8	186	123.2
103	23.6	145	74.0	187	124.4
104	24.8	146	75.2	188	125.6
105	26.0	147	76.4	189	126.8
106	27.2	148	77.6	190	128.0
107	28.4	149	78.8	191	129.2
108	29.6	150	80.0	192	130.4
109	30.8	151	81.2	193	131.6
110	32.0	152	82.4	194	132.8
111	33.2	153	83.6	195	134.0
112	34.4	154	84.8	196	135.2
113	35.6	155	86.0	197	136.4
114	36.8	156	87.2	198	137.6
115	38.0	157	88.4	199	138.8
116	39.2	158	89.6	200	140.0

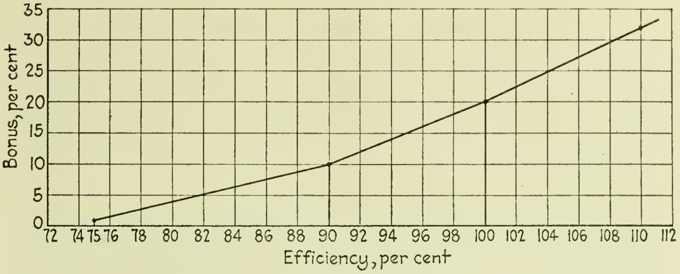
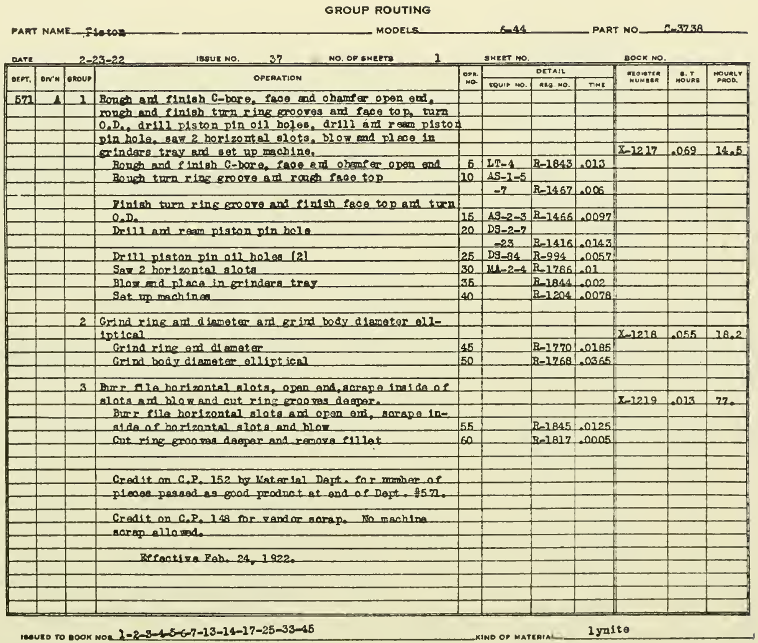


FIG. 1—CHART SHOWING THE RELATION BETWEEN EFFICIENCY AND THE PERCENTAGE OF BONUS

group. It thus always appears as one-man hours, independently of how many workers may subsequently be assigned to a designated group. All detail operation standard-times and the total group standards are listed on a shop routing for ready reference as shown in Fig. 2.

TABLE 2—EQUIVALENTS OF MINUTES IN DECIMAL HOURS

Min.	Equivalent Hours	Rate of Production per Hour	Min.	Equivalent Hours	Rate of Production per Hour	Min.	Equivalent Hours	Rate of Production per Hour
0.01	0.000167	6000.0	0.34	0.00567	176.5	0.68	0.01133	88.2
0.02	0.000333	3000.0	0.35	0.00583	171.4	0.69	0.01150	87.0
0.03	0.000500	2000.0	0.36	0.00600	166.7	0.70	0.01167	85.7
0.04	0.000667	1500.0	0.37	0.00617	162.2	0.71	0.01183	84.5
0.05	0.000833	1200.0	0.38	0.00633	157.9	0.72	0.01200	83.3
0.06	0.001000	1000.0	0.39	0.00650	153.8	0.73	0.01217	82.2
0.07	0.001167	857.1	0.40	0.00667	150.0	0.74	0.01233	81.1
0.08	0.001333	750.0	0.41	0.00683	146.3	0.75	0.01250	80.0
0.09	0.001500	666.7	0.42	0.00700	142.9	0.76	0.01267	78.9
0.10	0.001670	600.0	0.43	0.00717	139.5	0.77	0 01283	77.9
0.11	0.001830	545.5	0.44	0.00733	136.4	0.78	0.01300	76.9
0.12	0.002000	500.0	0.45	0.00750	133.3	0.79	0.01317	75.9
0.13	0.002170	461.5	0.46	0.00767	130.4	0.80	0.01333	75.0
0.14	0.002330	428.6	0.47	0.00783	127.7	0.81	0.01350	74.1
0.15	0.002500	400.0	0.48	0.00800	125.0	0.82	0.01367	73.2
0.16	0.002670	375.0	0.49	0.00817	122.5	0.83	0.01383	72.3
0.17	0.002830	352.9	0.50	0.00833	120.0	0.84	0.01400	71.4
0.18	0.003000	333.3	0.51	0.00850	117.6	0.85	0.01417	70.6
0.19	0.003170	315.8	0.52	0.00867	115.4	0.86	0.01433	69.8
0.20	0.003330	300.0	0.53	0.00883	113.2	0.87	0.01450	69.0
0.21	0.003500	285.7	0.54	0.00900	111.1	0.88	0.01467	68.2
0.22	0.003670	272.7	0.55	0.00917	109.1	0.89	0.01483	67.4
0.23	0.003830	260.9	0.56	0.00933	107.1	0.90	0.01500	66.7
0.24	0.004000	250.0	0.57	0.00950	105.3	0.91	0.01517	65.9
0.25	0.004170	240.0	0.58	0.00967	103.4	0.92	0.01533	65.2
0.26	0.004330	230.8	0.59	0.00983	101.7	0.93	0.01550	64.5
0.27	0.004500	222.2	0.60	0.01000	100.0	0.94	0.01567	63.8
0.28	0.004670	214.3	0.61	0.01017	98.4	0.95	0.01583	63.2
0.29	0.004830	206.9	0.62	0.01033	96.8	0.96	0.01600	62.5
0.30	0.005000	200.0	0.63	0.01050	95.2	0.97	0.01617	61.8
0.31	0.005170	193.5	0.64	0.01067	93.8	0.98	0.01633	61.2
0.32	0.005330	187.5	0.65	0.01083	92.3	0.99	0.01650	60.6
0.33	0.005500	181.8	0.66	0.01100	90.9	1.00	0.01667	60.0
			0.67	0.01117	89.6			



GROUP ROUTING

PART NAME Piston MODELS 6-44 PART NO. C-3738

DATE 2-23-22 ISSUE NO. 37 NO. OF SHEETS 1 SHEET NO. BOCK NO.

DEPT.	DIV'N	GROUP	OPERATION	OPR. NO.	DETAIL			REGISTER NUMBER	S. T HOURS	HOURLY PROD.
					EQUIP NO.	REG NO.	TIME			
571	A	1	Rough and finish C-bore, face and chamfer open end, rough and finish turn ring grooves and face top, turn O.D., drill piston pin oil holes, drill and ream piston pin hole, saw 2 horizontal slots, blow and place in grinders tray and set up machine.					X-1217	.069	14.5
			Rough and finish C-bore, face and chamfer open end	5	LT-4	R-1843	.013			
			Rough turn ring groove and rough face top	10	AS-1-5 -7	R-1467	.006			
			Finish turn ring groove and finish face top and turn O.D.	15	AS-2-3	R-1466	.0097			
			Drill and ream piston pin hole	20	DS-2-7 -23	R-1416	.0143			
			Drill piston pin oil holes (2)	25	DS-84	R-994	.0057			
			Saw 2 horizontal slots	30	MA-2-4	R-1786	.01			
			Blow and place in grinders tray	35		R-1844	.002			
			Set up machines	40		R-1204	.0078			
		2	Grind ring and diameter and grind body diameter elliptical					X-1218	.055	18.2
			Grind ring and diameter	45		R-1770	.0185			
			Grind body diameter elliptical	50		R-1768	.0365			
		3	Burr file horizontal slots, open end, scrape inside of slots and blow and cut ring grooves deeper.					X-1219	.013	77.
			Burr file horizontal slots and open end, scrape inside of horizontal slots and blow	55		R-1845	.0125			
			Cut ring grooves deeper and remove fillet	60		R-1817	.0005			
			Credit on C.P. 152 by Material Dept. for number of pieces passed as good product at end of Dept. #571.							
			Credit on C.P. 148 for vendor scrap. No machine scrap allowed.							
			Effective Feb. 24, 1922.							

ISSUED TO BOOK NOS. 1-2-3-4-5-6-7-13-14-17-25-33-45 KIND OF MATERIAL lynite

FIG. 2—TYPICAL ROUTING SHEET GIVING THE STANDARD TIME AND THE TOOL-GROUP STANDARDS FOR THE DETAIL OPERATIONS ON A PARTICULAR PART

APPLICATION OF GROUP STANDARD-TIME

To obtain a clear idea of the application of group standard-time and its use as a basis for a wage-incentive plan, let us consider a single production line, manufacturing and assembling pistons in an automobile-engine plant. This unit has its operations arranged in sequence, in what is known as a progressive line of manufacture. It starts with the rough-machining of the casting and ends with the finished product with piston-rings inserted. The production line includes both machine and bench-assembly operations. Parts move without a break from one operation to the next in a steady flow. Some operations may be done by a single workman; others may have several workers in parallel, depending on the volume of production and with what detail it is practicable to subdivide operations.

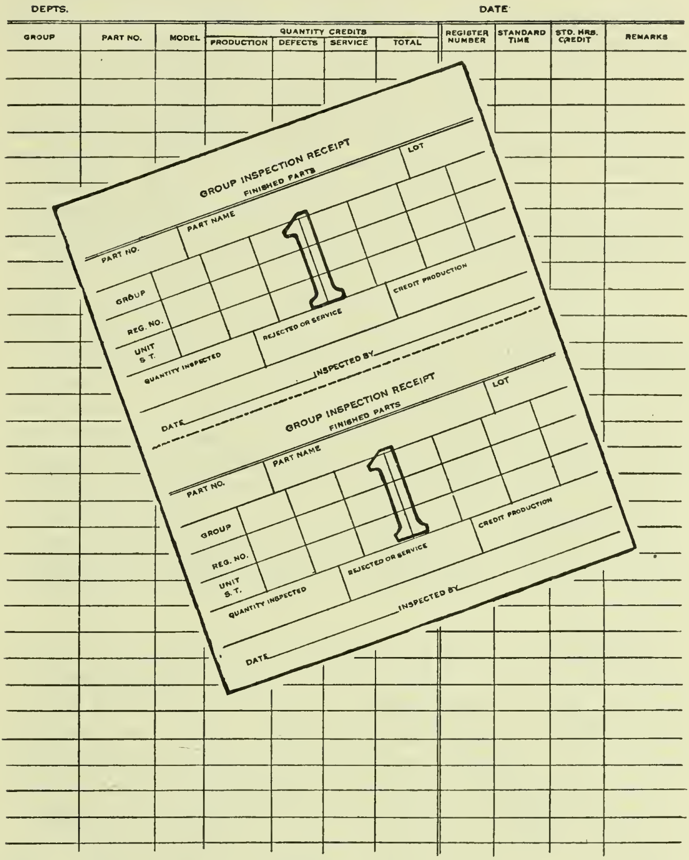
It is proposed to keep no check on quantities passing intermediate operations or on those completed by individual workers on operations in parallel but to give credit for finished pistons passing final inspection. This particular production line may be located in a department also producing connecting-rods or any other line of manufacture. Such a line embraces a production unit of the factory, and is technically known under the group plan as a *division.* Its workers are primarily interested in the production of pistons; they know little or nothing about conditions on the connecting-rod line and should not be grouped with it.

A study of the piston-line division indicates that there are two distinct classes of machine work, besides bench operations. The machine operations are lathe turning and grinding. There are thus three major operations, and the workers group themselves naturally into these three classes of work. It is thus seen that a division may consist of any number of distinct groups. They are bound together because they receive credit only for the finished product at the end of the division, while they are distinct and independent so far as efficiency measurement and bonus reward are concerned. A group is debited with the actual man-hours of the workers in the group and is credited with the group standard-hours multiplied by the number of pieces passing the division. That is, the quantity count is not taken at the group operations but at the end of the division of work. The ratio between standard hours and actual hours constitutes the efficiency measurement for which a bonus percentage is paid. For convenience, the entire pay-period of 1 or 2 weeks is run on a cumulative basis, and bonus is computed on the average efficiency attained for the period. Obviously, for a given product, the fewer actual man-hours there are in a group, the higher the efficiency of production will be and the greater the per cent of bonus earnings. The incentive is, therefore, to obtain a high rate of production per man per hour consistent with good quality of product.

Each member of a group receives the same percentage of bonus at the end of the pay period, but it is computed on his wage earned while assigned to the group; and the total amount earned by each worker will, therefore, depend on his hourly base-rate and number of hours worked. No job tickets are used. A shop timekeeper will handle from 300 to as high as 600 group workers. A list of employes is tabulated daily for each group, and elapsed time is taken at the end of the day from the entrance time-clock. If a worker is transferred out of or into a group, a transfer slip noting the time is recorded by the timekeeper and the elapsed hours are charged accordingly. Standard-hours credit is obtained from the finished inspection reports of quantities, multiplied by the group standard-time. The forms used in this connection are reproduced in Fig. 3. The method of tabulating group records daily in the time office, and of computing bonus, is indicated by the form shown in Fig. 4.

A group may produce various parts having different time-standards, and therefore different direct-labor costs and, since we use no individual elapsed-time job-tickets, labor costs are computed from the group cost per standard-hour. For given base-rates, the labor cost will be constant per standard-hour and per piece for all efficiencies above 100 per cent. If the average efficiency falls to 90 per cent, the labor cost will increase less than 2 per cent.

In the above discussion, we have assumed that the production line has its operations arranged in sequence. Such lines are, of course, ideal for the grouping of workers. But in the class of factory we are considering, there may be anywhere from 5 to 40 per cent of the direct-labor operations that cannot be arranged progressively. Such departments usually consist of miscellaneous small machines, automatics, grinders, or minor bench operations. However, it has been our practice up



DEPTS. DATE

GROUP	PART NO.	MODEL	QUANTITY CREDITS				REGISTER NUMBER	STANDARD TIME	STD. HRS. CREDIT	REMARKS
			PRODUCTION	DEFECTS	SERVICE	TOTAL				

GROUP INSPECTION RECEIPT
FINISHED PARTS
1
PART NO.
PART NAME
LOT
GROUP
REG. NO.
UNIT S. T.
QUANTITY INSPECTED
REJECTED OR SERVICE
CREDIT PRODUCTION
INSPECTED BY
DATE

GROUP INSPECTION RECEIPT
FINISHED PARTS
1
PART NO.
PART NAME
LOT
GROUP
REG. NO.
UNIT S. T.
QUANTITY INSPECTED
REJECTED OR SERVICE
CREDIT PRODUCTION
INSPECTED BY
DATE

FIG. 3—FORMS EMPLOYED IN CONNECTION WITH THE SECURING OF THE STANDARD-HOURS CREDIT FOR A GROUP OF OPERATORS

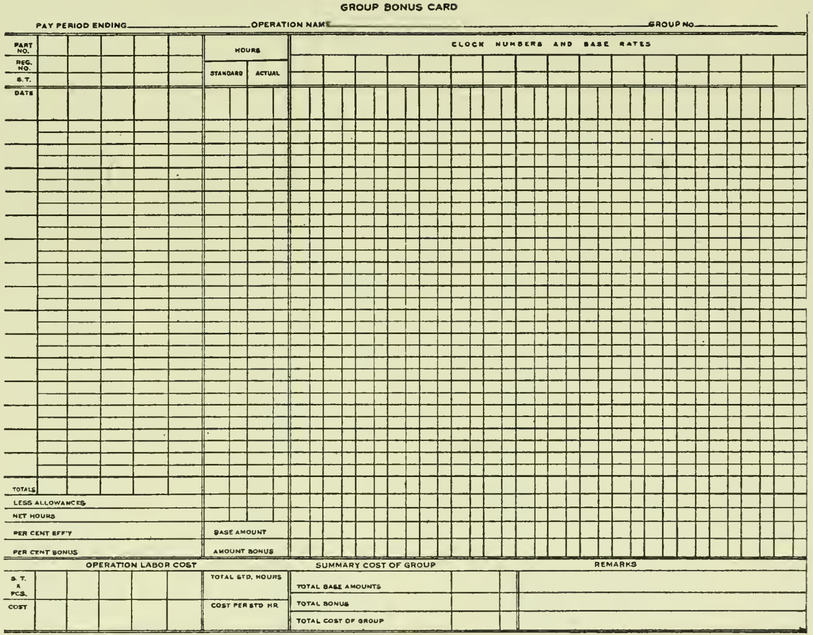
to the present to group such departments according to similar operations, being careful to retain the personal interest of the worker in the group production. Credit is given only for finished parts as they leave the department. So far, such grouping has been entirely successful. If there were any special departments of work in such factories that could not be grouped to advantage, there would be no objection to leaving them on an individual-incentive basis or even on a straight day-rate basis.

Indirect labor has been grouped extensively wherever a "community of interest" can be maintained between workers. They must have a common interest in the results of their own efforts. Storeroom labor, unloading materials from cars, boxing and loading automobiles for shipment and similar classes of work where the effort is measurable, have been grouped with very good results.

In summarizing our experience during the past 4 years with the group-bonus plan, it should be borne in mind that the particular feature involved is the principle of grouping employes and not the wage-incentive table that happened to be selected. Very likely any one of several incentive plans could have been used in connection with grouping and have produced satisfactory results. This one was selected because we thought it would be more adaptable to changing factory conditions than a system of fixed-group piece-rates having their value in dollars rather than in time. It also offers the same incentive to high production as could be obtained from piece rates.

Although the primary purpose of grouping was to simplify the factory system and to reduce the amount of clerical detail, it developed that there were many advantages from an operating standpoint. Much less material is tied-up in process, and full advantage can be taken of the mechanical arrangement of progressive lines whereby parts are made to flow from one operation to the next in a steady stream through single or parallel operations. Under the continuous-flow method of production, the checking of quantities after individual operations becomes difficult, if not impracticable; so, the logical method seems to be to count from the end of the line and credit groups of workers. Nearly every such production line has its "neck of the bottle" or several of them. If these can be speeded up, the whole line benefits.

One of the early advantages noted was the speeding-up, or elimination, of slow workers. Hence, it has been



GROUP BONUS CARD

PAY PERIOD ENDING________ OPERATION NAME________ GROUP NO.________

PART NO.
REG. NO.
S. T.
DATE

HOURS
STANDARD
ACTUAL

CLOCK NUMBERS AND BASE RATES

TOTALS
LESS ALLOWANCES
NET HOURS
PER CENT EFF'Y
PER CENT BONUS

BASE AMOUNT
AMOUNT BONUS

OPERATION LABOR COST
S. T. x PCS.
COST

TOTAL STD. HOURS
COST PER STD HR.

SUMMARY COST OF GROUP
TOTAL BASE AMOUNTS
TOTAL BONUS
TOTAL COST OF GROUP

REMARKS

FIG. 4—CARD ON WHICH THE GROUP RECORDS ARE TABULATED DAILY IN THE TIME OFFICE AND THE BONUS COMPUTED

our experience that more production per man-hour has been obtained under grouping than under a previous individual-incentive plan. From the viewpoint of factory operation it has meant the elimination of job tickets and elapsed-time records for group operations. This has meant saving in clerical detail. It has added to productive time, because there are always some delays if employes are required to keep count on quantities, obtain job tickets or furnish information to shop checkers.

Conditions Favorable to Grouping

It may be taken for granted that factory workers cannot be grouped indiscriminately, or under all sorts of conditions, with any hope of success. The individual incentive must not be lost sight of. If a group worker feels that others will carry his burden whether he exerts himself or not, there is no incentive for him to put forth his best efforts. Consequently, conditions must be such as will enable the individual to realize that the results are going to depend on his own interest in the work. Progressive lines of sequence operations, grouped according to major operations, are ideal for this purpose. On the other hand, where operations are not progressive but have a community interest, equally good results are obtained. As an example, a group of 20 men snagging miscellaneous castings on an average tonnage basis gave a remarkable performance because they had one interest, the output in pounds of castings per man per hour. Storeroom attendants, caring for and deliving materials, were successfully grouped by placing the incentive on the basis of hours time per 100 units produced by the factory.

The essentials for grouping are that workers must be placed close together as a team, and also be able to see the results of their own efforts. Operations should be largely repetitive. The results of each day's group-work should be made known as early as possible the following day to maintain the individual interest. The best results are obtained by tabulating this information cumulatively from day to day over a pay period of 1 to 2 weeks. It is essential that the computation of earnings be made so simple that the average factory worker can figure out his own earnings quickly, just as he would on straight day's-work. For this purpose a wage table, such as Table 3 but more in detail, is posted in the shop. Most employes use a copy of the bonus scale such as is shown

TABLE 3—WAGE TABLE FOR PRODUCTIVE WORKERS; COMPUTED FROM TABLE 1

Employe's Hourly Base Rate	Hourly Earnings at 85 to 130 Per Cent Efficiency									
	85	90	95	100	105	110	115	120	125	130
$0.25	$0.268	$0.275	$0.288	$0.300	$0.315	$0.330	$0.345	$0.360	$0.375	$0.390
0.26	0.278	0.286	0.299	0.312	0.328	0.343	0.359	0.374	0.390	0.406
0.28	0.300	0.308	0.322	0.336	0.353	0.370	0.386	0.403	0.420	0.437
0.30	0.321	0.330	0.345	0.360	0.378	0.396	0.414	0.432	0.450	0.468
0.32	0.342	0.352	0.368	0.384	0.403	0.422	0.442	0.461	0.480	0.499
0.34	0.364	0.374	0.391	0.408	0.428	0.449	0.469	0.490	0.510	0.530
0.35	0.375	0.385	0.403	0.420	0.441	0.462	0.483	0.504	0.525	0.546
0.36	0.385	0.396	0.414	0.432	0.454	0.475	0.497	0.518	0.540	0.562
0.38	0.407	0.418	0.437	0.456	0.479	0.502	0.524	0.547	0.570	0.593
0.40	0.428	0.440	0.460	0.480	0.504	0.528	0.552	0.576	0.600	0.624
0.42	0.449	0.462	0.483	0.504	0.529	0.554	0.580	0.605	0.630	0.655
0.44	0.471	0.484	0.506	0.528	0.554	0.581	0.607	0.634	0.660	0.686
0.45	0.482	0.495	0.518	0.540	0.567	0.594	0.621	0.648	0.675	0.702
0.46	0.492	0.506	0.529	0.552	0.580	0.607	0.635	0.662	0.690	0.718
0.48	0.514	0.528	0.552	0.576	0.605	0.634	0.662	0.691	0.720	0.749
0.50	0.535	0.550	0.575	0.600	0.630	0.660	0.690	0.720	0.750	0.780
0.52	0.556	0.572	0.598	0.624	0.655	0.686	0.718	0.749	0.780	0.811
0.54	0.578	0.594	0.621	0.648	0.680	0.713	0.745	0.778	0.810	0.842
0.55	0.589	0.605	0.633	0.660	0.693	0.726	0.759	0.792	0.825	0.858
0.56	0.599	0.616	0.644	0.672	0.706	0.739	0.773	0.806	0.840	0.874
0.58	0.621	0.638	0.667	0.696	0.731	0.766	0.800	0.835	0.870	0.905
0.60	0.642	0.660	0.690	0.720	0.756	0.792	0.828	0.864	0.900	0.936

in Table 1, and compute earnings directly from the average group efficiency.

SUMMARY

The group plan is primarily applicable to repetitive work arranged in progressive production lines of sequence operations, but it is also applicable to non-progressive operations where the individual interest can be centered on definite results per man-hour of labor. Like any other wage-incentive plan, it has for its main objective the speeding-up of the production rate per employe; that is, *intensive production.*

It is being used because it simplifies the factory detail where a high rate of production is desired on repetitive work, and where a wage incentive is employed as the means of obtaining intensive production. Its application has gained favor among factory executives because it speeds-up production as compared with the individual plan on the same work, ties up less material in process and minimizes clerical detail in the factory.

To have a group plan remain in successful operation over an indefinite period, it is necessary to maintain the interest of the individual worker in the group effort. It must be simple for computation of individual earnings, flexible so as to meet changing factory conditions and easily understood by each group member.

THE DISCUSSION

CHAIRMAN H. W. ALDEN:—What effect has this group-bonus wage-incentive plan had on the inspection problem in the plant? Has it decreased the ratio of inspectors to producers and improved the general quality and output of the product?

A. J. SCAIFE:—How is the individual rewarded in each group? How does the reward of the man old in the service compare with that of the man just employed, and how do the rewards of the efficient and the inefficient workers differ?

JAMES H. MARKS:—What procedure is followed with respect to his bonus in the case of a man who is discharged or quits during the pay period?

We also have installed a group-bonus system in our plant that is very similar to that of Mr. Wennerlund, although I believe that our groups as a whole are somewhat larger than his, as we have grouped entire departments. The basis of bonus that we use is certain standards of efficiency that we have determined for machine operators, assembly men, molders, steam-hammer men and others. We have set various standards of efficiency rather than use one of 80 per cent as described by Mr. Wennerlund. We offer a 1-per cent bonus for each point that the group efficiency exceeds the predetermined standard of efficiency. We take care to have varying rates per hour for the different employes in the group, in proportion to their speed and ability. What is meant when the efficiency of the group exceeds 100 per cent?

S. W. TAYLOR:—I worked for Mr. Wennerlund for a number of years. We based our time-studies on 100-per cent production for the good average worker.

J. C. WILLIAM SMITH:—At the Willys-Overland plant in Toledo we use a similar plan, but it is straight piece-work and each man in the group is assigned a specific piece-price that is paid him for each operation he performs. For example, in our gear-housing group there are eight Gisholt machines, together with numerous other machines; if the group turns out 500 housings at the end of the line, for the day, we pay each man in the group for 500 housings. Where there is more than one machine on the same operation, as in the case of the eight Gisholts, the pay is divided equally among the eight men, if agreeable to them. If not, each man is paid for the number of housings on which he performed the operation, but the total paid for does not exceed 500.

The foreman makes the decision as to the number, if the men do not agree. In this way we have done away entirely with the job tickets mentioned by Mr. Wennerlund. The method obviates paying for the gear housings before they are actually received. It gives us the individual efficiency of each operator and reduces the number of our inspectors, checkers and truckers, as the work does not touch the floor. It demonstrates to each man that he is paid exactly for the work performed. We use guaranteed piece-work prices and the price is not changed before the end of a year.

W. G. CAREINS:—I am connected with the Milwaukee plant of the Nash Motors Co. We use group-bonus plans. We use some straight piece-work and, in fact, we are not partial to any one system, but use what we think is best adapted to the particular line of work in hand. In case a new man comes into a group to perform an operation that requires some time for him to become proficient in it and to develop a standard rate of production, such as the instance cited of the Gisholt machines, we would pay him the day rate for a certain length of time, perhaps 1 week, 10 days or 3 days, depending upon the length of time needed for him to become proficient. The other men in the group take the balance of the pay and split it among themselves.

In some departments where we use the group-bonus plan to provide an incentive to different classes of labor, we have what we call Class A, Class B and Class C men who take different proportions of the total bonus-price. We find that this works very satisfactorily; for instance, in engine tests, where a certain number of men probably handle the hoists, another class of men who are real engine mechanics do the class of work they are fitted for and a third group run the engines. In our Kenosha, Wis., plant, every operation is individual piece-work. This plant has never gone into the bonus plan or the group plan in any way.

H. K. REINOEHL:—Must the group be kept small so that the individual members can watch each other for the purpose of eliminating the poor producer?

E. BOUTON:—In a group-bonus plan, how are costs maintained level? There is a fluctuation of cost. When does Mr. Wennerlund arrive at a true cost, or an actual cost?

J. L. LANNEN:—What disposal is made of scrap?

J. A. MURRAY:—We have in the General Motors Truck

Co. plant the group-bonus system, which was installed under Mr. Wennerlund's supervision. We have been operating this system for about 2½ years and I cannot say enough in its praise. It relieves me of a great amount of detail that we had to work out when operating under the individual-effort system and has reduced considerably the amount of clerical work in both our time-keeping and production departments.

We have approximately 525 machines at work on the group-bonus plan and they are split into 23 groups. Each group is really a plant in itself, inasmuch as there is a machine in each group or division for each operation and the work flows down the line progressively, passing only from machine to machine and never jumping across from one line to another.

We also maintain what we call a day-work well. This is built up when we install a line of machines or route new parts over the line. It naturally follows that during the trying-out of the group and the demonstration of tools, a number of parts are made that we pay for on the day-work basis. This number of hours is always held back from the operators and constitutes what we call a day-work well. We consider this well the same as a machine, the company owning it, inasmuch as we already have paid for the work done on the day-work basis. Our aim is to pay a bonus only for the parts actually produced on bonus. This well, as you will see, needs to be built up only once, and we maintain it until a group is either done away with or a new well is established.

E. KARL WENNERLUND:—The effect of the group-bonus plan upon inspection methods is that inspection must become more exacting as soon as an incentive plan is inaugurated; however, that is true of every plan. For example, the highest type of car might be built on straight day-work and some inspection would be necessary; but if that same car were built under a wage-incentive plan it would be done to speed-up production. Just as soon as production is speeded-up, the tendency, at least, is for the quality of the product to deteriorate. I do not wish, however, to convey the impression that the group-bonus system alone would cause this tendency but, with the factory organization remaining the same, additional inspectors are needed to counteract it.

In regard to inspection under the group-bonus system as compared with the inspection of individual piece-work, we have gradually developed certain practices. Under either plan there is the same inherent tendency to speed-up production and for quality to go down; so every piece or a certain percentage of pieces must be inspected in any event. Inspection is not eliminated nor quality improved simply by introducing groups, but we are coming to the point where we pay for the good product only, unless it can be shown that the material from the manufacturer was defective or had defects when it came to the department. The result is that the group tries to turn out better work instead of trying only to get the work through.

For example, in a plant where we are introducing the group-bonus plan now, armatures for starting systems are tested and a certain percentage of them fail to pass inspection. It is very difficult to say that this is the fault of any one employe. Formerly, the armatures were laid aside and, when a certain number of them had accumulated, they would be sent through on a day-work ticket to be fixed. It was not practicable to deduct the cost from anybody's pay, because it was impossible to determine who caused the trouble. We told these employes that we would pay only for the good armatures and that we had found the defective ones to constitute about 2 per cent of the total number. Therefore, we said we would add 2 per cent to the time standard, so that, if the defects ran higher than that, there would be a fine against the men. On the other hand, if this proved to be unfair, we would adjust it. Now, when a defective armature comes through, the tendency is for the inspector to throw it back into the line, and we do not have to get the elapsed time for fixing it. We used to get time tickets that were charged to expense.

Defective products fall into one of two classes; the first is defective work that can be corrected, and the other is simply junk. If the product has to be scrapped, this is done in the same way as under the individual-effort system. If it can be fixed on the line, it is sent back without any ticket and without any credits. That plan has had a very good effect in standardizing our defective work.

As to the division of earnings, there may be a man in the group on an operation that requires great skill, the result of years of service, and he may have to work to limits that are within 0.0005 in. In the same group there may be a boy who drills little pin-holes on a vertical drilling machine. He places the piece in a jig and could not spoil it if he tried. Naturally, the skilful man should not have the same earnings as the boy. In this particular case we paid the skilled man a base rate of 60 cents per hr. and the boy 30 cents per hr. The efficiency of that group at the end of the pay period might be 110-per cent production. If so, each member of the group would get a 32-per cent bonus on his earnings. If each worked 100 hr., the mechanic would get $60 flat rate with 32 per cent added and the boy would get $30 flat rate and 32 per cent additional. That plan takes care of all of our division of earnings. Sometimes we have girls working on a small bench-operation and we may have a skilled mechanic on the very next operation; it would not be fair to them or to any one to divide their earnings equally.

The case of workers who were discharged, or who quit, was a little troublesome at first. We used to tell them we would give them straight pay and that at the end of the pay period we would give them the bonus they had earned. Sometimes such men left and did not know where they were going, so that plan did not work out very satisfactorily. Furthermore, our State law is rather strict; it requires that, if a man is discharged, he shall be paid that same day. We adopted a very good working rule by taking the average efficiency of the group up to the end of the previous day, and figuring the bonus on that basis for the workman who leaves the service. For example, suppose the pay period to have extended 5 days and that the workman quits today at noon. He might be discharged; if so, we are compelled to pay him his actual flat-rate multiplied by the hours he has worked, the same as for straight day-work. But we are under a moral obligation, at least, to pay him a bonus; so we take the efficiency up to the previous night. It may be 90 or 100 per cent, and it may not be the same as at the end of the pay period but, so far as the company is concerned, it follows the law of averages, being sometimes a little over and sometimes a little under. It seems to be entirely satisfactory to our men, and we have had no complaints. A man seems to think it is entirely fair if he gets his straight pay plus the bonus up to the last night on the basis of average efficiency; so it has worked out very simply.

Concerning the size of a group, we must all concede that the individual-effort incentive is right, and we find that we must secure the individual interest of the employe in his own effort. He must be placed right next to his team-mates and on similar work. In isolated cases one can do almost anything, but certain fundamental principles apply to groups and one cannot get very far away from them.

We started a complicated job recently at one of our large factories that employs about 2000 people. The first operation was performed by about eight girls each of whom did exactly the same thing, picking up little bars and measuring them with micrometer calipers. The next major operation was assembling, done by probably 8 to 10 persons, who put a bushing in and fastened it. The next group, 8 to 10 men, did the machine work. It would be a mistake to put all of these operatives in one gang or one group. The girls on the first operation knew nothing about the machine work. Assuming that the group became reduced from eight to seven because of one who quit, seven operatives would get the pay of eight. If this amount were spread out among a lot of other operatives, the group would not get the benefit of it; so, we try to keep our groups fairly small, to localize the interest. When we wanted to group the girls that performed the first operation, they objected on the ground that two of them were slow operators; so, the slow ones were replaced by faster workers.

The primary object of the group-bonus plan is to reduce the amount of clerical work. We found later that this is only a minor consideration. The big effect is in increasing production because the slow workers are eliminated, and those who remain help each other out on certain operations, all along the line.

As to what is meant when the group efficiency exceeds 100 per cent, the question should be: "What do you mean by 100 per cent?" It means whatever meaning is given it. The standard time for an operation is the time taken repeatedly by the average competent workman, over an extended period and eliminating lost and waste time. It is not the time in which the operation can be done by an expert. The efficiency measurement is the standard time divided by the actual time, and 100 per cent on production is the average time made by the average competent workman. For example, a good workman ought to dig a certain number of feet of ditch in a day; another man may dig only half as much and a third man may dig 25 per cent more; the efficiency measurement is from an assumed standard that we called "standard time."

In regard to new men that enter the group, we found it a good plan to carry a new man for 3 days and at the end of that period to throw him automatically into the group unless the superintendent asks specifically to have the probation time extended. The effect is that the company carries the new man for 3 days and the men in the group carry him for the following 3 days; after that it is up to the man to make-good in the group.

Another method which, in principle, amounts to the same thing, is to charge half of the new man's time to the group and the other half to the expense account and not to allow the man to participate in the bonus while he is on that basis. The effect is that the group carries only half of the new man's time, and he does not get the bonus and therefore tries to enter the group. That works out very well.

The question regarding the proper size of a group and why it should be small has to do with individual effort. The worker should be able to see that the output depends upon his own efforts. As an illustration, we had 20 men snagging castings. Handling this work on an individual basis was very difficult because we had to weigh all the castings for each man and keep track of them. To handle them on a day-work basis would have required about 30 instead of 20 men; so, we placed them all in one group and took the foundry report for a certain number of pounds per day, and allowed a certain number of man-hours per 1000 lb. That plan worked well. We dispensed with from six to eight men that had been required under the individual-effort system. The only reason we could get the men to work under this plan was that they had an incentive to increase the output. If we had tried to group them with the molders, for instance, we would not have succeeded. The plan demands very close cooperation; the men must be near together so they can see that every man's efforts count. A certain output from a group costs us a certain amount of money and, if we are making only one part, the cost of the group operation is the money we pay divided by the number of pieces we obtain. That is positive and actual. We may have various operations going through the group. We may have a heat-treating group in which there are as many as 80 different parts, ranging all the way from small washers to front axles. There may be 20 to 30 men, split into various groups that perform the different operations. We can determine the cost because every one of those pieces has its own standard time. The time in which a front axle goes through is known to be a certain number of minutes or tenths of hour and, if a washer has to be hardened, that would represent a smaller unit of time. When the pay period closes we know certain positive facts. We know the amount of money paid out and the number of pieces produced and we have a standard time-value on each one. We determine the total number of actual man-hours in a group and the total number of standard man-hours and can therefore compute the cost per standard hour. For example, if each standard hour costs us 54 cents and the standard time on a certain product is 0.1 hr., the cost will be $0.054. Above 100 per cent, the cost is uniform. If the efficiency runs down to 90 per cent, the bonus decreases 10 per cent and our costs increase about 1.5 per cent. If the efficiency is reduced to 50 per cent, the cost increases; but, whatever the cost is, it can be computed.

We have very uniform costs. Under the group-bonus plan the cost efficiency is maintained above 90 and 100 per cent fairly well, and we get a uniform cost for given base-rates. No matter how many or how few pieces go through, or how many dividends there are, we can always determine the costs.

The general superintendent of an important factory that has been using the group-bonus system for about 5 months wrote me as follows:

> The results shown already are certainly most gratifying to us and substantiate our belief that the complete installation of this system in our factory will result in a wonderful saving to our company. A rough average of the results disclosed about a 23-per cent reduction in the number of employes, compared with the old method, 20-per cent increased real earnings for employes, and 49-per cent increased man-hour production.
>
> As you are aware, we installed this system a small group at a time, explaining to the employes its advantages to them and to the company, and in no instance have we had any difficulty in throwing-in additional groups. We are frank to say that the company and the employes are sold 100 per cent on this system, and the factory in general is benefited under this new method of working, as it is more quiet and orderly than under the old method. We hope to complete the necessary details and to have all operations working under this system by the first of the year.

To show why these men are taking-up this group-bonus system, I will include an extract from a letter written by the general manager of a company that builds a well-known car. He says:

> After years of experience, I am of the opinion that the group-bonus system is the best system adaptable not only to progressive manufacturing, but to other classes of work where the measurement of individual production would be expensive or burdensome. We are using the group-bonus plan because we feel that it is better adapted to our needs than the former individual-effort system, and that it is much more satisfactory than the usual piece-work system to both the management and our employes. Our experience during the past year has confirmed our views on this subject.

Another extract from an unsolicited letter by the general superintendent and the chief executive officer in one of the largest automobile factories in this Country is as follows:

> In regard to the group-bonus wage-incentive system, one of the first benefits is a doing away with a large number of clerks and carriers that were necessary under the old method of payment. We have found also that we have less labor trouble. I believe this is because the men take more interest in their work. We have found also that we are getting more production where we have this system thoroughly worked out. Another item of interest is the fact that we have reduced the percentage of scrap.

This last factory was formerly 100 per cent on the individual-effort system and had been ever since its start. It had worked on the individual-effort system for years. It was turned over to a group-bonus plan a few years ago and is still operating that way.

I wish to show that the factory managers who have taken this up and have tried it are with us on the proposition. I have received other letters along the same line. Unless these men were with us and found the group-bonus plan to be the right thing, there would be no use trying to put it through. It succeeds because each one who tackles it right and gives it proper cooperation, becomes one of our good friends and boosters.

A METHOD OF DEVELOPING AIRCRAFT ENGINES[1]

By Capt George E A Hallett U S A[2]

The general method of procedure taken by the Air Service before beginning the actual design and construction of the necessary types of aircraft engine is outlined and the four steps of the development subsequent to a very complete study of existing domestic and foreign engines are stated.

After checking over the layouts, if all the details are agreed upon by both the designer and the Engineering Division, the contract is placed, usually for two experimental engines, and the construction work is begun.

Acceptance tests are made to demonstrate that the engine is capable of running at normal speed and firing on all cylinders. These are followed by the standard performance test made on the dynamometer at McCook Field. The results of the latter test determine whether the engine can enter the 50-hr. endurance test. The engine is then torn-down and inspected for wear. Suggested modifications are embodied in reconstructed engines which eventually fulfill the requirements. Descriptions of the various tests are given and commented upon.

The development of an aircraft engine is defined as the work which is done between the time of making the initial scale layout and the time when the engine is ready for production. The purpose of this paper is to describe briefly the method pursued by the Engineering Division at McCook Field in accomplishing this work. Before entering into a description of the methods of aircraft engine development the need for the many types will be explained briefly and what are the general steps to be taken before beginning the actual design and construction of these types.

At the close of the war, very few fully developed types of engine were available to our Air Service. We had a bombing engine which was good for various two-seater airplanes, but we did not have entirely satisfactory pursuit, training or heavy-bombing engines. The operations group of the Air Service promptly prepared a set of performance specifications for 16 types of aircraft for as many military uses. Working from these specifications, a similar list was prepared of the sizes and types of engine that would be needed in this program, with the idea of making them available as rapidly as possible; and a very complete study of existing domestic and foreign engines was begun. First, standard test methods were laid down. Next, standard test-report outlines and contents were decided upon. Then domestic and foreign engines were tested in two ways, (*a*) by a so-called standard test that included thorough and accurate measurements of all phases of the engine's performance and in some cases, (*b*) by a 50-hr. test for the purpose of comparing the durability of the engine with that of others. As this work got well under way, we started to work out improvements for existing domestic types and arranged with suitable contractors for the design and construction of new types that appeared to be needed. In a very few cases where no contractors could be interested the designs were laid down at McCook Field.

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[1] Semi-Annual Meeting paper.

[2] S.M.S.A.E.—Chief of powerplant section, engineering division, Air Service, Dayton, Ohio.

The study of existing domestic and foreign engines provided valuable data on such characteristics as brake mean effective pressure, horsepower per cubic inch of piston displacement, pounds of engine per cubic inch of displacement, comparative freedom from vibration of different types and comparative durability. It was noted that the fuel consumption was controlled largely by the compression-ratio, the type of carbureter and the type of manifold, provided the engine showed reasonable mechanical efficiency and brake mean effective pressure. These data on the characteristics of the best existing engines served as a fairly good guide in regard to what types of design and construction would prove most satisfactory for incorporation in the engines to be developed, and it should be noted particularly that since all these measurements were taken on the same standard equipment and by the same standard methods, the results were unusually comparable.

DEVELOPMENT PROCEDURE

In general, the steps in the development of engine types have been as follows:

(1) The design and layout are started in most cases by civilian contractors
(2) The Engineering Division, upon receipt of these

layouts, generally checks the stress and weight calculations

(3) The engine is checked from a standardization and installation point of view; in other words, the gasoline, oil, water, throttle and electrical connections must be standard and located at points that will be accessible after the engine has been installed; provision must be made for mounting the standard electric starter, gun synchronizers, gasoline pump, tachometer drive and electric generator. Much serious trouble in airplane design and engine maintenance is thus avoided

(4) The layout is checked to see that it has embodied the latest experience of the Engineering Division and its contractors in design and constructional details

The Engineering Division acts as a clearing-house for approved practice in design and construction. For example, the good points of an engine designed by the Engineering Division or a civilian company can be applied to an engine designed by some other company. This practice has been of great benefit when engines have been

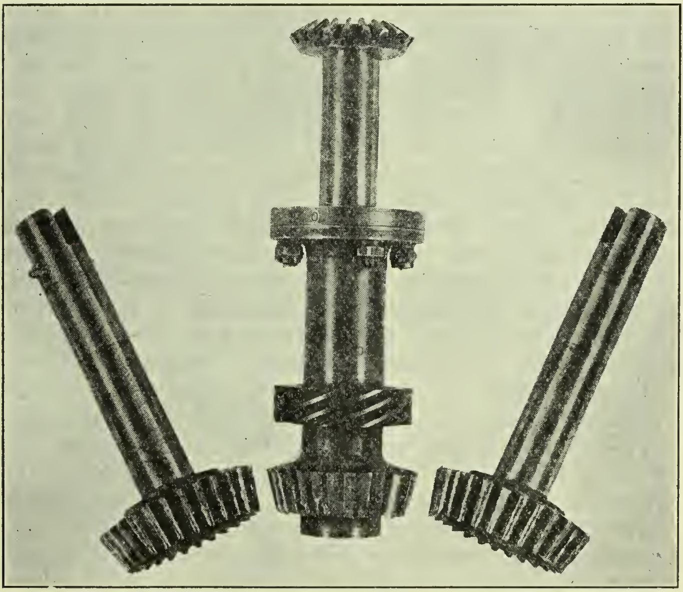

FIG. 1—TYPICAL SET OF CRANKSHAFT GEARS THAT HAVE BEEN DAMAGED BY WHIPPING OF THE CRANKSHAFT

designed by the various civilian contractors as well as when designed by the Engineering Division.

After checking the layouts and when all the details have been settled to the satisfaction of the designer and the Engineering Division, the experimental contract is placed and the construction work started. The contract generally calls for at least two engines, inasmuch as it has been found that much delay is avoided by having an extra engine ready for test in case of an accident occurring to the first one; however, when the design is considered unusually experimental, the contract generally is made for one engine with a rather large allowance for spare parts, which amounts almost to a second engine unassembled. After the construction of the engine is started, the detailed inspection generally is carried on by the contractor, but usually the plant is visited and the work looked over during the most interesting stages of construction.

ACCEPTANCE TESTS

After the engine has been completed, the acceptance test is run, sometimes at McCook Field and sometimes at the contractor's plant, according to where the work can be done most conveniently. The nature of these tests is generally little more than a demonstration that the engine is capable of running at normal speed and firing on all cylinders. The more experimental or nonconventional the engine is, the less severe are the tests. This is due to an attitude on our part of being responsible for the design, since it has been checked so carefully from the start and since many basic changes are often incorporated. The acceptance test generally shows whether the engine has been constructed and assembled in a satisfactory manner and the engine is accepted or rejected on the basis of the results of this test. It undergoes the standard performance test at McCook Field.

If the cylinder construction of the new engine is not of a well-proved type, the construction of the complete engine usually is held up pending the construction of one cylinder, the making of thorough tests of it on one of the universal test engines. These single-cylinder tests often show many troubles but, fortunately, most of them can be remedied by modifying the design or methods of construction. Frequently changes in the design of the valves, valve-guides or valve-gear are found necessary. Sometimes the compression-ratio proves too high or unneces-

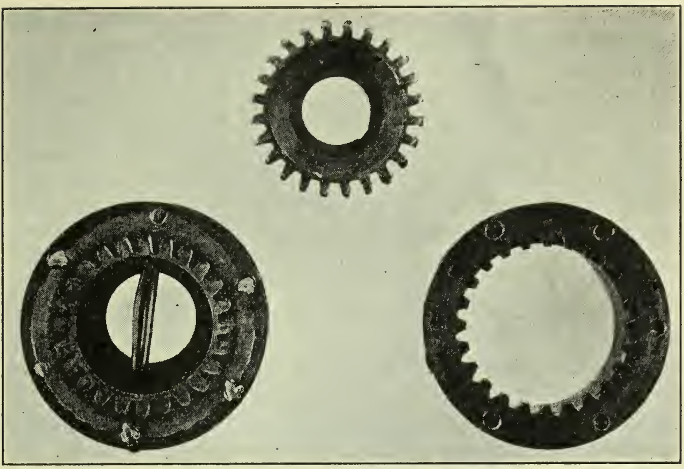

Fig. 2—Magneto Coupling in Which the Rivets Have Been Sheared and the Splines Worn as a Result of the Torsional Vibration Transmitted from the Crankshaft

sarily low, and the best location and number of spark-plugs as well as the proper spark-timing and piston clearance have to be ascertained. If the results with the single cylinder should prove too bad, the entire project would be held up indefinitely pending the design and development of a more satisfactory cylinder; or perhaps abandoned. Generally the results can be made satisfactory and the work proceeds.

We find that the single-cylinder tests are very satisfactory in everything but fuel-consumption. Fuel-consumption readings are comparable only with other single-cylinder results and are not comparable with results to be obtained from multi-cylinder engines. The writer is responsible for the use of these single-cylinder engines for this work and is endeavoring to carry the idea of testing "by units" much farther. In newly designed engines, the fuel-pumps, oil-pumps, water-pumps, ignition systems and carbureters can all be thoroughly tested in an accessories laboratory entirely independently of the rest of the engine. In one of the engines designed at McCook Field an entire gearcase assembly was tested to determine whether the complicated lubricating system was satisfactory. This was done by mounting it in the accessories laboratory and driving it at normal speed by an electric motor, using a thin oil of a viscosity similar to that of the Air-Service specification oil when at the working temperature. Faults in the oiling system were found much more quickly and cheaply in this way than by testing the assembly on a complete engine. Obviously, if enough parts of the engine can be tested separately at small expense and risk, the complete engine is far more certain of satisfactory results when assembled, and many costly wrecks can be avoided. It is believed that much more progress can be made along this line.

The Standard Test

When the completed engine is placed on the dynamometer at McCook Field ready for its standard performance tests, assuming that it has not been placed in good running condition previously, the procedure is about as follows: The engine is motored over with plenty of good oil and water circulating through it until the horsepower needed to motor it over at normal speed bears a predetermined relation to the horsepower to be expected from the engine, thus assuring satisfactory and standard conditions within the engine. Carbureter chokes are selected that will produce a satisfactory manifold depression when the engine is being motored over at normal speed. We do not deceive ourselves by running engines with large chokes and low manifold-depressions that are impracticable in flight; therefore, the manifold vacuum is held between 1 and 2 in. of mercury. After this the jets can be adjusted so that they meter an amount of fuel in accordance with that to be expected per horsepower-hour, thus insuring a carbureter setting and mixture-ratio easily within the running range. Further adjustments must be made by actually running the engine under its own power, but the preliminary steps avoid much guessing at the start. Many other precautions are taken at this stage of the work but they are covered in detail in Engineering Division reports.[3]

As soon as the engine is running satisfactorily, steps are taken to improve its performance, if possible, and it is then put through the standard test planned to provide all the desired data on the performance of the engine. These include the obtaining of full-throttle-power and propeller-load-power curves, which give a good indication of the fuel consumption in throttle flight; measuring the fuel and oil consumption; and the data needed by the airplane designer such as rate of water flow, water-tem-

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[3] See Engineering Division reports Nos. 1506 and 1507.

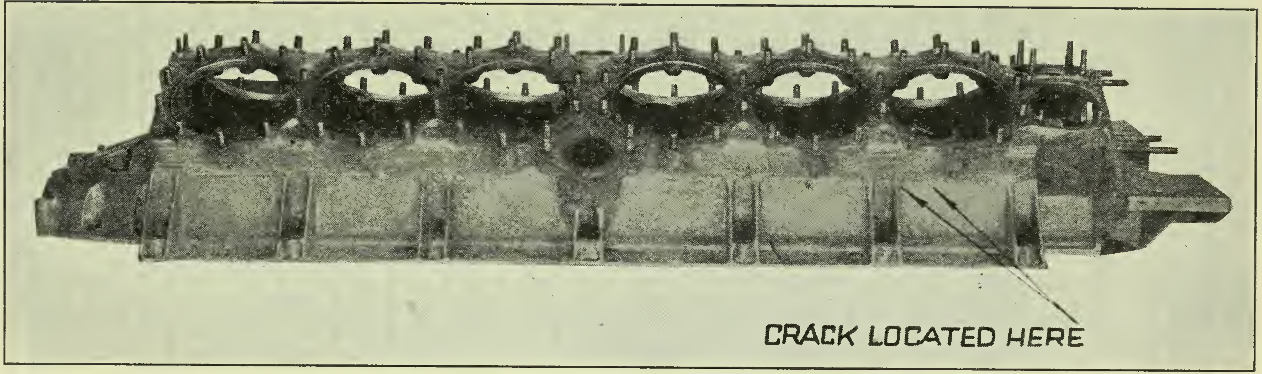
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FIG. 3—CRANKCASE THAT CRACKED IN SEVERAL PLACES WHILE UNDERGOING TESTS

perature rise and the like. In case of failure of the engine during the standard test, parts that are affected are studied carefully and, if practicable, they are replaced and the test is continued. Fortunately, most of the engines we have developed have been able to get through the standard tests and at least get started on the long endurance test that follows. At the close of the standard test the engine is dismantled and inspected with the idea of ascertaining whether it is in shape, or can be put in shape, to start on the endurance test. If at the end of the standard test the engine is not in shape to start on the endurance test, it may be repaired, redesigned or abandoned, according to the results shown in the standard test.

The 50-hr. test is a standard method of comparing the durability of engines. It is a first step in developing an engine toward satisfactory durability. These tests are nearly always made on a torque-stand with a propeller providing the load, and are based on the normal speed and power of the engine. A propeller is selected that will permit the engine to run at its normal speed with the throttle wide-open and absorb the normal rated power of the engine under these conditions. The 50-hr. test consists of 10 non-stop runs of 5-hr. duration, each run beginning with ½ hr. at full throttle and full speed with the propeller as described above and continuing for the remaining 4½ hr. when throttled down to the speed at which the propeller absorbs 9/10 of the power at full throttle. Due to the fact that in an airplane the engine generally is run for only 5 or 10 min. at anything approaching full throttle and is then throttled to a much lower speed when the airplane is taken up to an altitude at which the power output is greatly reduced because of the decreased density of the atmosphere, the 50-hr. test is much more severe than 50 hr. of flying; in fact, many Liberty engines flying as many as 180 hr. before overhaul are not in as bad shape as the same engine would be at the end of a 50-hr. test.

During the 50-hr. tests the gasoline and oil consumption and power fluctuations are noted carefully and the valves are watched for heating; in fact, the entire engine is watched carefully and an accurate record is kept of all troubles and of work that becomes necessary. Many troubles are encountered due to torsional or other periodic vibrations in the crankshafts, especially in those which are over four throws in length. These vibrations result in the breakage of the gears used in driving camshafts or magnetos, of the magneto couplings and even of the magnetos themselves. Sometimes these troubles require several modifications before they are eliminated completely.

Fig. 1 shows a typical set of camshaft drive-gears that have suffered from such "crank-whip." A Hispano-type magneto-coupling in which the rivets have been sheared and the splines worn, due solely to torsional vibration transmitted from the crankshaft, is illustrated in Fig. 2. Fig. 3 shows a crankcase that was found to be cracked in several places, one of which is indicated by the arrows. A portion of a similar crankcase modified to overcome this weakness is illustrated in Fig. 4. Fig. 5 shows some connecting-rod bearings that have failed partly, due to the bad load distribution which was caused by unsatisfactory connecting-rod design. The bearings out of the same engine after another 50-hr. run with connecting-rods of a new design appear in Fig. 6. As will be noticed by comparing Figs. 5 and 6, the bearings in the latter show great improvement. Bearing No. 5 was injured by the breakage of a defective crankshaft. Fig. 7 illustrates some exhaust-valves that have suffered excessive wear on their tips and stems and from burning. In this case all these troubles were traceable to a defective

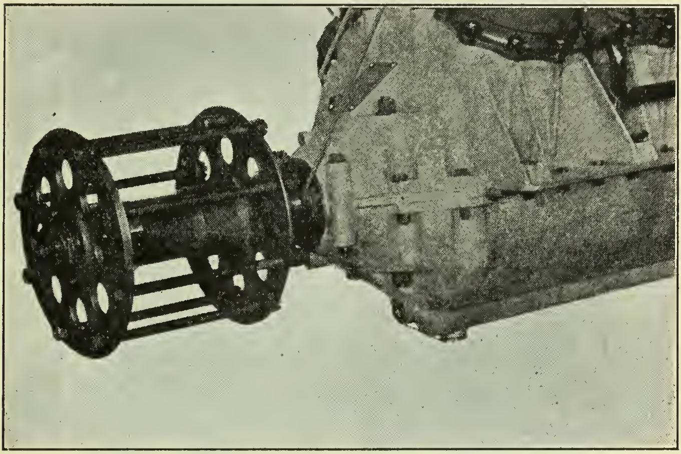

Fig. 4—Portion of a Crankcase in Which the Design of the Ribs in the Upper Half Was Modified To Provide Additional Stiffness

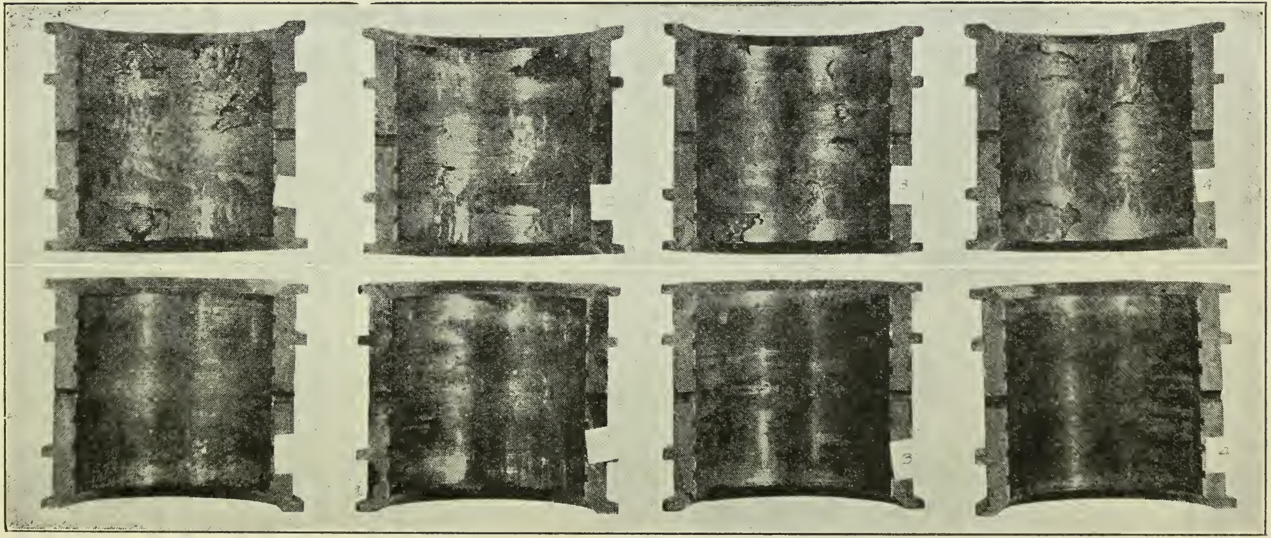

FIG. 5—FAULTY DISTRIBUTION OF THE LOAD CAUSED THE FAILURE OF THESE CONNECTING-ROD BEARINGS

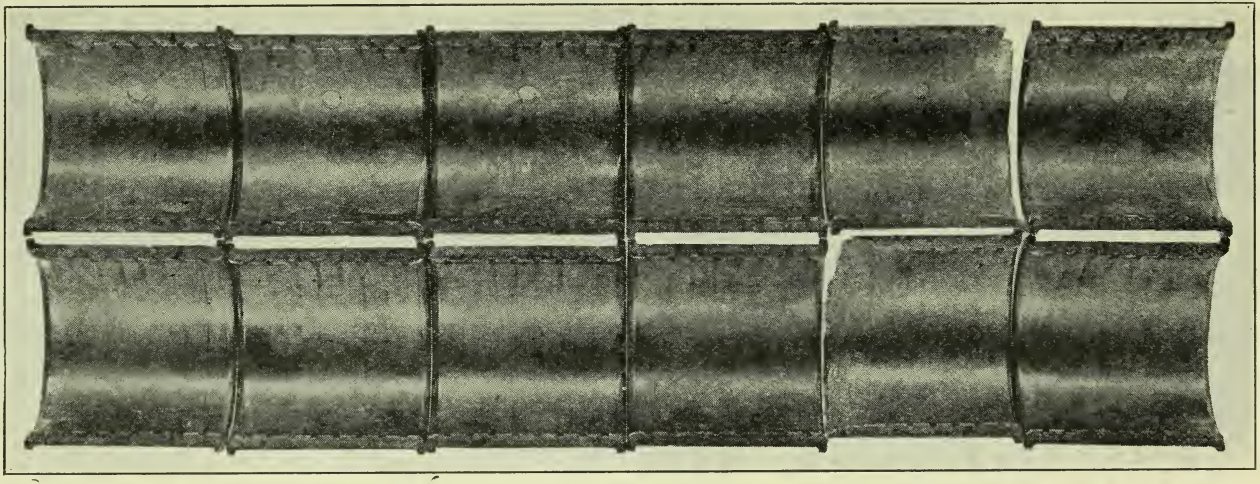

FIG. 6—AN IMPROVED FORM OF CONNECTING-ROD BEARING

rocker-arm design that imposed too much side-thrust on the valve-stems, an extreme instance of this type being illustrated in Fig. 8. Fig. 9 shows the valve-stem-guide bushings and Fig. 10 is reproduced from a photograph of the valves from the same engine after a 50-hr. run with a modified valve-gear that overcame the difficulty.

Some engines prove hard on spark-plugs, crack their water-jackets under the severe testing conditions, burn their valves and break minor parts such as valve springs, rocker-arms, oil lines and the like. Occasionally we have a serious wreck before the completion of the 50-hr. test; but in every case we endeavor to get 50 hr. of running if we possibly can, since this is not an acceptance test but a method of obtaining 50 hr. of a standard kind of wear on an engine so that its durability can be compared with that of other engines.

TEAR-DOWN AND INSPECTION

After the 50-hr. test, the engine is torn-down and inspected carefully. Most parts are inspected both before and after cleaning. All parts are removed to the photographic room where they can be inspected more accurately and can be photographed conveniently. The party that inspects the torn-down engine generally consists of the writer, a designing engineer, the test engineer assigned to the engine in question, the mechanic who has had the engine in charge and other interested persons. As they are examined the parts are checked off systematically from a printed list and the condition and appearance of each part is passed upon. We find that the appearance of parts means much. An agreement concerning each piece is usually reached before passing to the next one.

The help of the Materials Section at McCook Field has proved invaluable in arriving at the reasons for breakage and other kinds of material troubles. A decision as to what will be done about improving the part is withheld until after the receipt of the Materials Section's report. The results of the inspection or conference on the dismantled engine are noted and an agreement is reached with the designer or builder of the engine as to what should be done to meet the criticisms and recommendations. If the results warrant it, arrangements are made with the constructor to build a modified engine in accordance with the recommendations made after the last inspection. We are glad to say that the engine, when re-

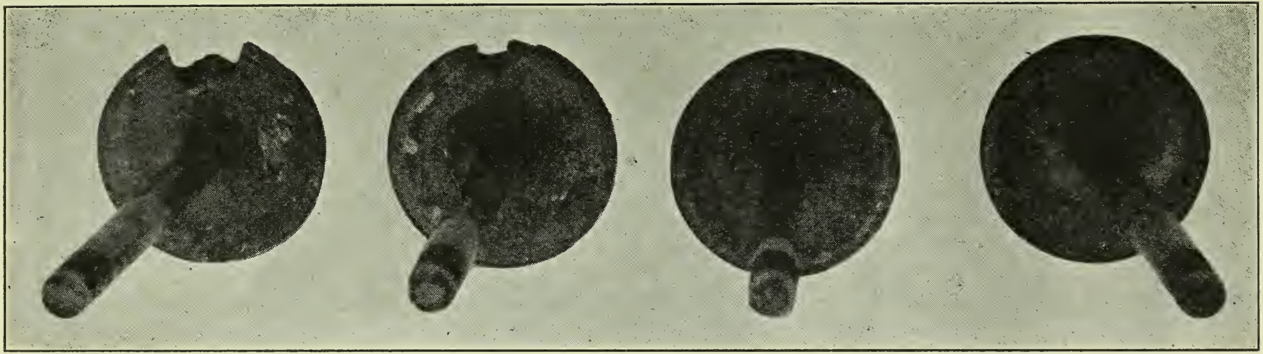

Fig. 7—Exhaust-Valves That Wore Excessively on Their Tips as a Result of a Defective Rocker-Arm Design

constructed, accepted and given a check-run and the 50-hr. test, generally completes the test in a satisfactory manner. It is then torn-down again and inspected. When the need of additional improvements becomes apparent, they usually are made.

The photographs of parts of the first engine that were taken while the engine was torn-down at the end of the 50-hr. test are exceedingly useful for making comparisons with the observed condition of the second engine, inasmuch as many different engines are handled by our organization and we may not carry in mind perfectly the conditions found in the engines from one test to another. After a sample engine has gone through one of these 50-hr. tests satisfactorily, it generally is considered good enough to warrant the purchase of 8 or 10 more for experimental use.

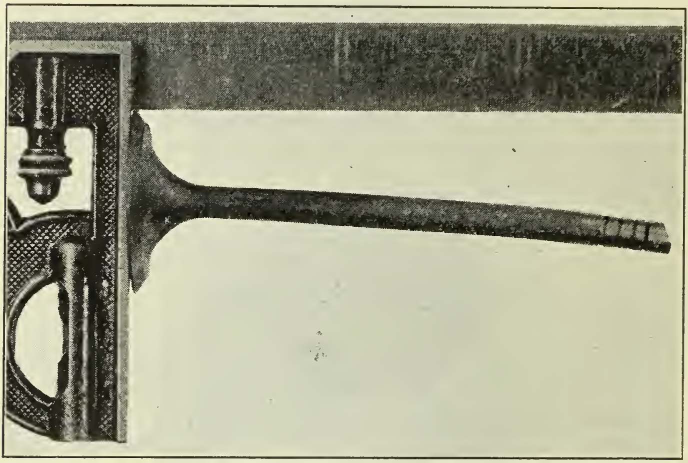

FIG. 8—AN EXTREME INSTANCE OF THE ILL EFFECT OF THE SIDE-THRUST THAT WAS IMPOSED UPON AN EXHAUST-VALVE DUE TO IMPROPER DESIGN OF THE ROCKER-ARM

FLIGHT AND SERVICE TESTS

The first of these engines is put into flight test as promptly as possible, because usually an additional crop of troubles shows up when a new engine gets into the air for the first time. Such troubles as improper venting of the carbureter float-chamber may develop in flight because of changes in the relative pressures in the float-chamber and the throat of the carbureter while the airplane is

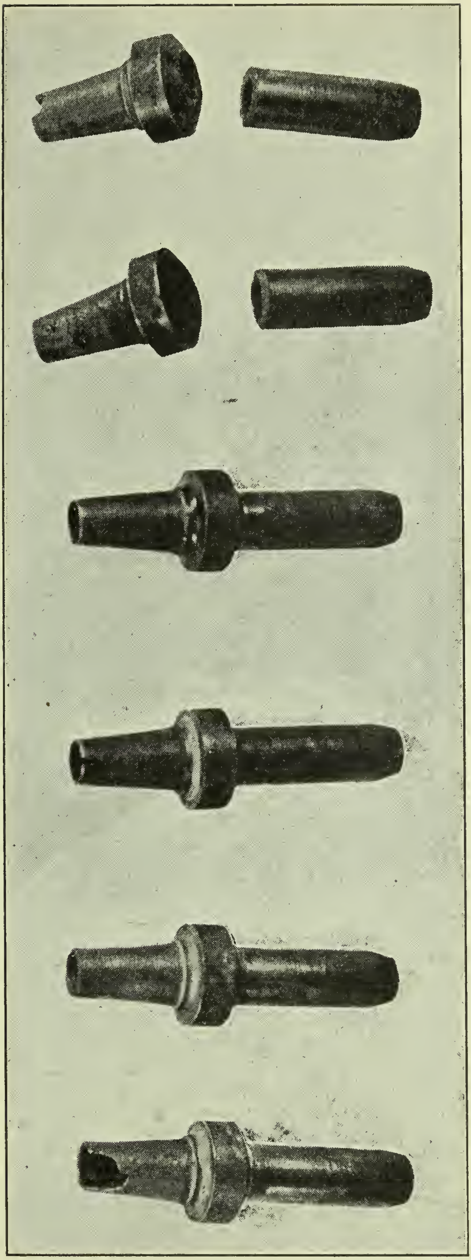

FIG. 9—VALVE-STEM-GUIDE BUSHINGS TAKEN FROM AN ENGINE AFTER IT HAD BEEN TESTED AT MCCOOK FIELD

moving at high speed. The distribution of the mixture may prove bad in sharp maneuvers, even though it may be satisfactory in normal flight. Occasionally, troubles in the water-circulation system appear when the airplane is in abnormal positions. Propeller-hub troubles are encountered occasionally, but the most troublesome point is vibration. Engines that seem to run smoothly on the test-block will shake badly in flight and cause trouble in the surrounding structure. Sometimes the accessibility of an engine does not prove to be as good after it is installed as it seemed to be on paper and on the test-block.

When an engine has reached the point where it has passed the 50-hr. test and is working satisfactorily in the air, and a few others of the same type are in use in the Air Service, it is time to try much longer tests of 100, 200 or even 300 hr. duration. These are of the nature of refining. The remaining weak points which did not show up in the 50-hr. test will begin to appear in the longer test and generally can be corrected then. It usually is possible to develop the engine so that it can run at higher speeds, higher compression-ratios or higher brake mean effective pressures, thereby effectively reducing the specific weight of the engine without increasing the difficulties of production or the cost, and at the same time insuring better fuel economy.

When a new type of engine is sent to some field where it has been unknown, many troubles may be encountered which are due to the fact that the personnel is unfamiliar with that particular engine. These might not have shown up during the tests at McCook Field. The maintenance conditions and methods at such a field are likely to be different from those where the engine was developed, and troubles of this nature may easily arise. Then, too, there seems to be a particular species of trouble that never occurs until the airplane gets out of sight of the airdrome. For example, parts will work loose that had never shown any signs of doing so on the dynamometer or block tests or even during the short flights around the airdrome. Another class of trouble is that due to the psychology of the pilot; in other words, to his lack of confidence in the new engine.

In the minds of the pilots, no engine is likely to be considered durable and reliable until it has gone through an experience of quantity production and wide use similar to that of the Liberty engine. Practically all engines

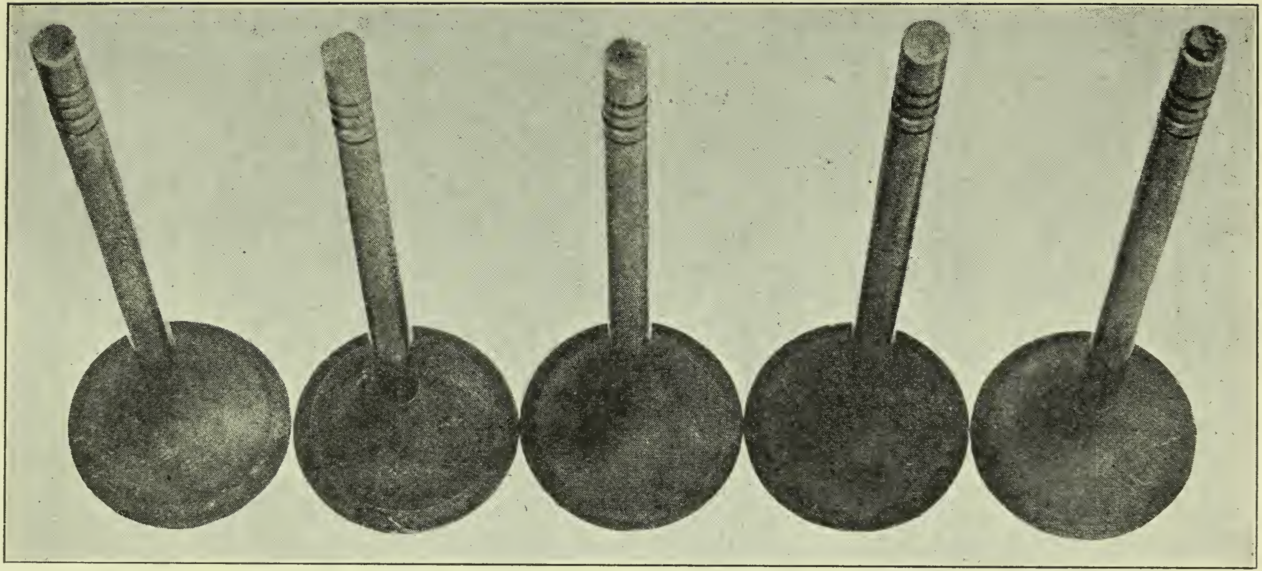

FIG. 10—VALVES OF THE SAME ENGINE, FROM WHICH THE BUSHINGS SHOWN IN FIG. 9 WERE TAKEN, AFTER IT HAD BEEN SUBJECTED TO A 50-HR. TEST WITH A MODIFIED VALVE-GEAR

have had their troubles when first put into use, and the Liberty engine is no exception. We are informed that due to its more hurried development the first 250 Liberty engines developed cracks in their crankcases; that the first 750 had light crankshafts, which occasionally broke, and weak connecting-rods. Since these weaknesses were rectified the engine has had such wide and satisfactory use that the pilots have long ago forgotten that phase of its development, and it now enjoys their greatest confidence.

In addition to developing complete engines, much work has been done in perfecting accessories that are not really parts of the basic engine but are fairly general in their application, such as carbureters, spark-plugs, piston-rings, valves and pistons. The results of this work have been applied to whatever engines could be benefited and have produced considerable standardization as well as improved performance. Between January 1919 and the present time development has been started on 23 engines. Of these two which were failures have been abandoned. Sixteen have been successes, and five which are not yet complete give promise of being successful.

THE DISCUSSION

PROF. E. P. WARNER:—The paper prepared by Captain Hallett and read by Mr. Jones seems to me to be peculiarly important, first directly because of its explanation as to the method of developing engines at McCook Field, where they are certainly doing more development work than at any other agency in the Country, and second because we who are interested in aeronautics should have a clear understanding of what that method is and have a viewpoint of our own on what the best procedure in distributing the work of development between the Government and private enterprise is. I think that in the next few months, or in the near future at least, we are likely to see a strong drive from some quarters for a revision, in one direction or another, of the present method.

I have heard from various quarters in the industry the sentiment expressed that the money that goes into the Air Service in general is a good appropriation gone wrong, and that it would have been better to turn the same money over to the manufacturers to keep alive on. On the other hand, we sometimes find the view on the part of those connected with the Government work, or strongly sympathetic to the Government work, that the manufacturer should confine his attention in peace times to producing those military supplies that are turned over to him by a Government agency, which, in the case of the Air Service, would mean that all the research in engine design should have been handled by McCook Field. It is very important that such research should continue, and it was handled very ably during the war; but at the same time there is danger in going too far in one direction or another. Owing to the departure of some of those in the automobile industry, I think the chances now are rather strong that those who remain may be attracted to the Government work more on the airplane than the engine side, and that the design will fall more and more into Government hands, a condition that would be dangerous in case an expansion of production facilities again became necessary.

If we get all the engineers, particularly those who are handling design, into the laboratories and debar the factories and designing staffs from going ahead with original work, we will be in a very poor position to expand, as bad as we would be if we allowed the factories to do their own work and did not enforce the exchange of results in research and original investigation. It is desirable, I think, that the information come into hands from which and through which it will be, as far as possible, impartially distributed to those who can make use of it for the development of aeronautics in general and for the good of the industry as a whole. Take a hypothetical case: If we suppose that all Government agencies such as that at McCook Field were to cease their research, it would be very difficult to get into the hands of a small company trying to produce engines or airplanes, perhaps more particularly airplanes, the results of the work that is being done by others. I am a very strong advocate of Government research and of the maintenance of Government laboratories, not only for research of its own but for the collecting of information on a uniform and impartial basis, and for the distribution of research data from whatever source they may come.

H. M. CRANE:—I have had a pretty long experience with private and with Government development work, and with the two in combination. I have known Colonel Bane for a long time, and I think that he has worked out about as just a balance between the advantages of the two systems separately as it is possible to work out.

We must not overlook the fact that it is desirable to have an educated personnel in the civilian outside field. It is equally important not to overlook the fact that it is necessary and desirable to have a highly educated personnel in the Air Service, in both the Army and the Navy, that will be the nucleus for an expansion during war. All of us who had anything to do with the early supply of war material know what a handicap it was to deal with men in the Air Service who knew nothing about the work. At present, however, we have at McCook Field enough men of very great air experience, who are thoroughly acquainted with all the difficult problems of engine development, are sympathetic toward the difficulties of such work and know the length of time it takes and the importance of it.

When I first had anything to do with aviation-engine design, the question of development did not mean anything, any more than it did with airplanes. The airplane designer seemed to be expected to stand or fall on the first airplane he produced of an entirely new design; if it did not work, the attitude was to throw it away and make an entirely new and completely different design. But it has been proved by experience that the best airplanes and the best engines cannot be obtained in that way. Mr. Jones will undoubtedly tell you that the best engines in use today are the ones that have had the longest period of development; and the ones that started soonest with any sound basis on which to go, and have been most consistently developed, are still the most useful engines.

I was told by a man in Washington who should have known better that you could not keep up with engine design unless you lived in France; that the designs changed so quickly and were improved so rapidly that, unless you were right next to the front, there was no opportunity to keep up at all. As a matter of fact, if you look over the standard engines used in the Air Service in this Country now, you will find that all of them are based on designs that were made from 1915 to 1917. I cannot at the moment think of any service engine in regular use today in the Air Service, that was designed originally at any later time. In other words, the 4 or 5 years of development of a sound, original design has far outstripped anything new that could be produced in a year or two, based on all the knowledge of that development work.

ELMER A. SPERRY:—I think there is a general feeling in engineering circles that in Colonel Bane's organization at McCook Field we have a remarkable agency, paralleled probably by no other one in the world, where aid is extended not only to the recognized designers and to those that have been longest in service and whose product has been longest under study and has been perfected gradually and persistently, but where the radical designer also has a wonderful reception and is not only encouraged and sympathized with, but given every possible aid. I have known of photographs being taken especially for designers so that they could have the benefit of the very latest developments in the particular line in which their endeavor lay.

It is, I think, beyond the knowledge of the lay engineer how completely Colonel Bane's wonderful organization has been developed, how he inspires his men, extending this even to the workers outside, and how broad and perfectly just it appears to those who have to deal with it. We now learn again from Captain Hallett's paper how extreme performance is not demanded in early attempts and how the designs are treated sympathetically, working toward the proposition of determining first the soundness and ultimate utility of the principle. I think this stands in rare contrast to the treatment that some designers have received from other quarters. I, for one, think that everything should be done to strengthen the organization at McCook Field. It is in good hands, the appropriations there in all probability reach farther and it accomplishes more than any similar organization. There is no organization in Europe that has attained the accuracy of bomb-dropping, the perfection in photography, and the ceiling that McCook Field has attained; or that has been as bold and progressive in a great many lines and as willing and daring to try even extremely radical things. In some cases these are successful and in every case they are extremely useful in teaching us what not to do, which every engineer knows is almost equally as valuable as finding out what to do. I think that as a body we should thank Colonel Bane and his whole organization to the very best of our ability and extend to them every support.

PROFESSOR WARNER:—Mr. Crane expressed my views perhaps better than I could myself, relative to the balance of justness and the accuracy of the balance that is being struck by the McCook Field organization at present. I hope I was not understood as offering any criticism of that or of any other present organization. I said rather that before any change is made in either of the two directions I suggested, we should consider very carefully what the desirable change is.

What I desired to emphasize mostly is the necessity of the development work, the necessity of sympathy between the Government and the industry and of an educated personnel in the industry in the development of a new type. I know of several airplanes used during the war that, according to records which we have received since the war, were very unsatisfactory when first brought out and did not really become satisfactory airplanes for pursuit use, although they were later among the best on the front, until they were modified and rebuilt 20 to 30 times.

If airplanes are to be designed by the manufacturer, if he is to do anything except produce them, it is necessary that he be in a position to undertake some of that modification. Of course, the present situation in the airplane industry is very unfortunate. I think Chairman Clark will bear me out in saying that the ideal condition would be one in which an order for an airplane, when given to a manufacturer, could be carried through by that manufacturer to the point where he considered the machine as satisfactory; in which he could do a large part of the experimental work and then put the machine out with a guaranteed performance and declare that it was satisfactory to him, instead of, as at present, having to receive an order for an airplane that has never been built, on the basis of design alone, and to submit that airplane which, as a rule, never has been flown before it gets into the hands of the Government.

That is the one point where we might perhaps hope for a better balance in the experimental work. It is a point regarding which it is very difficult to see any practical improvement at present. None of the manufacturers of airplanes is able to undertake that experimental work under present conditions; it has to remain in the hands of the McCook Field personnel. Some difficulty to the manufacturer in keeping in touch with the changes made and in incorporating what proves desirable in later design is likely to result. As has often been the case, a manufacturer may have to change from a pursuit machine to a day-bomber and then back to a ground-attack type, instead of taking a pursuit machine and being able to develop it himself, carrying it through half a dozen constructions, as has often proved desirable. It is necessarily difficult for the designer to profit fully by the results of experience with his designs in flight.

I think that is the one point in which a change in the present system might be made profitably. Recent practice in Government work has been somewhat in the direction of allowing the designer more freedom in profiting by the results of experiments on his productions, in allowing him to continue to build other machines that incorporate improvements suggested as the result of trials of first types.

MR. CRANE:—Professor Warner should not have thought that I was attempting to criticise his remarks. I was only trying to amplify them. However, his later remarks are along a line that is also extremely close to my heart. I think that his ideas are being worked out at McCook Field now as well as is feasible under our system of handling appropriations by Congress. There are a great many builders who would like to work at double their present prices; in other words, they would like to take a guaranty contract if they could get an appropriation large enough to pay for two or three failures before they came through with the final result. But under the scrutiny given all McCook Field contracts by more or less interested parties, it has seemed to be impossible, I understand, to give contracts on that basis. It is a much better basis without a doubt.

The size of this Country and the limited personnel that is available now in the airplane plants result in very serious lost motion between the designing and the development testing. The testing is bound to cover very considerable periods of time and it is impossible for the designer to spend adequate time at McCook Field when his factory is somewhere else; he has to be at home and active on his other work. This forces him more or less to take the printed or typewritten word on the results of tests, or information given him verbally after the tests have taken place. This prevents his seeing the actual results and is undoubtedly a very serious handicap.

The other handicap, an endeavor to overcome which was made during the war, is the year-to-year appropriation that makes it difficult to give a long-term development-contract to anybody with any certainty of it being supported through to the end. I do not know whether we can obviate those difficulties with our present system of Government, but I do know that in the last 2 years McCook Field has succeeded in overcoming one by one very many of the worst of the difficulties. They are not overcome entirely yet, by any means, and it probably will be impossible to do so under peace-time conditions.

FREDERICK E. MOSKOVICS:—It seems to me that there is something to guard against in Professor Warner's suggestion regarding the latitude to be allowed to the individual manufacturer for experimenting. I think there is much to be done in educating the individual manufacturer who is experimenting as to just what he should do. I recall vividly an incident that took place at McCook Field during the war, when one of these experimenters brought a new airplane there for a flight test. When the McCook Field authorities proposed to sand-load the wings and test it, the airplane representative immediately went to Washington to protest to a certain senator that McCook Field was going to "bust up his plane." They sent a commission down there; the Military Affairs Committee of the Senate took the matter up and almost peremptory orders were sent down there to let this fellow fly his machine without sand-loading. He was supposed to have done a certain amount of experimenting. Glenn Martin and others talked for hours with the pilot who was to fly the machine and begged him not to. We all talked with him and said it was absolutely sure death. The machine did everything in the air that the McCook Field authorities had predicted it would do, and the pilot was killed within 3 min. after the plane left the ground. There is a perfect illustration of letting a jackass do his own experimenting, without some check upon him.

MR. SPERRY:—I do not believe in too much private experimenting with airplane engines, using Government funds. I can understand the lure of doing one's own experimenting, but where is the superman who can produce results comparable with those that are produced when the same machine is taken to McCook Field for test, where it is brought immediately under the observation of 20 experts instead of one. These can look at it from as many different standpoints and can give advice for improvement along many lines, thereby tremendously accelerating the development. I firmly believe that the system that is being pursued has very many advantages, and that it should be continued until some better method has been found.

PROFESSOR WARNER:—The solution for preventing the type of accident that Mr. Moskovics mentions seems to lie in having a trained personnel in the industry; the securing of engineers who know something about their work before they are turned loose to experiment. I think no one would advocate letting anybody who wants to do so build an airplane and then have the Government fly it; but I do think that the industry ought to be encouraged to build up and keep up that organization of aeronautical engineers that was started during the war, and to secure trained men who are capable of conducting experiments. McCook Field certainly ought to continue; it would be absolutely disastrous if it were to cease its work of sand-loading the wings and of making investigations of airplanes. However, there are some things we cannot calculate. The maneuverability of airplanes is one of those things. No one appears to know really at present whether an airplane will be maneuverable or not, when it is started. The only way you can tell is by trying it out. For example, in the case of the machines that I spoke of, where it was necessary to destroy and rebuild 20 to 30 times before a satisfactory machine was secured, it was necessary to rebuild these machines to improve things that could not be calculated in advance.

It is a legitimate thing, it seems to me, for experiments to be conducted by the manufacturer, provided he has a competent, trained personnel of his own. You are dealing with a very difficult thing when you are trying to build an airplane at points 1000 or 2000 miles apart; there is a great amount of lost motion in calculating the tests on planes that have been redesigned.

MR. MOSKOVICS:—The point that Professor Warner brings up, goes without saying; but who is to determine the capabilities of this experienced designer? As to the ability of a manufacturer to do a certain amount of experimenting, I was impressed recently at Port Washington while watching a flight of a Loening seaplane, the first monoplane seaplane Mr. Loening built in regular production. The machine was developed by Mr. Loening's experiments.

If McCook Field could keep in intimate contact with other research fields, I would say that nothing should be left undone that could be done in that line and that it would be a great step in the right direction.

PROFESSOR WARNER:—Mr. Moskovics has, I think, an excellent illustration of the value of experimenting with a competent personnel. The man who is constructing for the Government should be authorized to do Government work. As to the agency to select competent workers, I think McCook Field is in a better position to say whether or not it would be safe to fly the machines. The authorities there are the most competent judges.

Mr. Crane:—I think McCook Field found during 1918 and 1919 very clearly what they were up against; that the outside designers would take an order for a fighting machine like the pursuit-plane and not try to consider what a pursuit-plane is for. They took a certain set of written rules and went ahead. The main idea in those days was that if you got speed enough and a rapid enough climb, that would secure an order. At that time there was no way of getting a really good all-round fighting machine in this Country, because there was no designer in the field outside who knew what it was or seemed to care what it was.

All of the early machines that came to McCook Field were lacking in every detail of a good field-machine except possibly speed and climb. The engines were not accessible; it was impossible to do the most ordinary work on them; the machines were difficult to set up and take care of, and in most cases you could not fire a gun successfully out of any one of them, which of course was the final and limiting objection. Since that time McCook Field has been more and more willing to work with the designer who is willing to try to work out their problems for them. They are striking the happy medium between discouraging the outside designer who simply has something to sell, or a weird idea that has murderous possibilities, and the other designer who will really aid them in their work. They have gradually modified their system of contracting until it is about as good, I think, as it can be at present.

Mr. Moskovics mentioned the Loening organization. I know the form of contract it has taken in recent years. The contracts have been of a continuing nature; that is, for four or five machines of a given type, no two of them alike, or ordered for and delivered at the same time. This has given an opportunity to modify each machine from the results of the one preceding.

To revert to war times, a special dispensation was made in the case of the Loening organization on sand-loading. I think that had never been done before. But McCook Field very wisely provided the first sand-loading test on the wings, with the idea that those wings were an entirely experimental set and should be tested properly under the supervision of the designer and his men. The wings stood up well, but one metal fitting crumpled up, which would have caused a serious accident in the air. But the deflection of the wings that occurred during different stages of the sand-loading under the supervision of the designer permitted him to modify the design and obtain a factor of safety something like 15 per cent higher, with no added material at all. It was a point that could not have been covered in previous calculations.

What we are doing is part of the education of the outside personnel, and it is a very important part. The airplane is the toughest problem we have today, for the reason that it is an assembled machine, and a machine in which the correct assembly is of the utmost importance. Our best motor cars are practically designed completely and constructed in individual plants. The assembled cars are never in the same class in efficiency. The airplane is still an assembled proposition. In designing a new airplane for the Government, one result is that we cannot reasonably expect the installation to be good on the first one. The attempt to take the time required to work out the installation to the last little details on a drawing-board, on a machine as complicated as an airplane, would be time wasted. The most important thing on the first design is to have it strong enough, and to fly it and determine that the general type is suitable, that it has the basic fundamental requirements that justify the development expense. When that has been done, by preliminary experimenting and preliminary sand-testing, and with those samples on hand that you can measure up and on which you can shift parts about, you can determine the desirable modification in structure.

Any other system than that is bound to result in a junk-pile that is all out of proportion to what we ought to have, or else a line of service machines that are really entirely unserviceable compared to what they might be under other conditions.

AIRPLANE PERFORMANCE FORMULAS[1]

By Edward P Warner[2]

Aerodynamic analysis relates mainly to questions of performance and stability, the latter including both maneuverability and control, but the designer's problems concern chiefly the prediction of the best possible performance. Accurate analysis, which would include a summation of the elemental resistances of an aircraft part by part and the making of many corrections, supplemented by tests of models in a wind-tunnel, involves much labor and expense.

When a preliminary choice of dimensions and specifications for a new type of an airplane is to be made or there is a question of the performance attainable with a given load and power, a shorter method becomes necessary. This is to be found in the derivation of simplified formulas and graphs.

The author illustrates by examples the process of deriving these formulas and considers in turn such elements as minimum and maximum speeds; climbing ability and the conditions under which an airplane will have a zero ceiling, that is, the limiting conditions under which flight is possible; rate and angle of climb, the latter controlling the possibility of getting out of a small field over a barrier; and the best ceiling possible. Under these heads he takes up such questions as fineness, which is the ratio of the parasite resistance to the wing area; and lift, weight and power and their relations in determining the various coefficients that are used. Values obtained theoretically from the formulas were checked by comparison with those of about 50 airplanes of various types of which the measurements and performance are known and application is made to specific examples. Numerous curves show the relation of the various coefficients that enter into the design and performance of the airplane.

The aerodynamic analysis of airplanes and that portion of airplane design dealing with the practical applications of aerodynamic theory divide naturally under the two heads of performance and stability, stability being taken to include maneuverability and control. A large share of the designer's problems, therefore, pertains to the prediction of performance and to the determination of a design that will give the best possible performance.

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[1] Semi-Annual Meeting paper.

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The analysis of performance may be carried out by several methods, the more refined and accurate of which involve much labor. To obtain a really accurate estimate of the speed or climb of an aircraft necessitates the summation of the elemental resistances part by part and the making of many corrections for such factors as the slipstream, the inclination of the thrust axis, interference between the parts of the structure, and the like. The labor involved, while justified in a final calculation, is too great to be undertaken lightly when preliminary estimates of performance are to be made or when a number of different possible arrangements of the parts of a machine are to be compared with respect to their aerodynamic qualities.

A much simpler device for approximating performance involves the use of a wind-tunnel test of a complete model. This is less accurate than the method of direct calculation, as the wind-tunnel model is so small that it is impossible to make it a true scale reproduction of a full-sized machine in every particular. A multitude of such items as fittings and stranded wires, small individually but of importance in the mass, are therefore omitted from consideration. Even if it were possible to include such parts, the scale effect would be so enormous that the results obtained by the usual method of scaling in proportion to the square of a linear dimension and to the square of the speed would give incorrect results. A wing made to 1/20 scale may logically be expected to give very nearly 1/400 the lift and drag of the full-sized wing at the same speed, but the results that would be obtained by attempting to make a 1/20-in. scale model of a cable ⅛ in. in diameter or of a honeycomb radiator would be almost meaningless and certainly would not repay the vast amount of effort involved. Furthermore, the wind-tunnel does not allow for the making of a correction for slipstream effect, a very important factor in the resistance of an airplane. On the whole, it is rather remarkable that notwithstanding these obvious and manifold disadvantages it is still possible to secure a reasonably close approximation to the performance of an airplane by a single test of the lift and drag of a complete model in the wind-tunnel. The results for maximum speed are somewhat better than are those dealing with climb, as the slipstream effect, ignored in the wind-tunnel test, is of relatively more importance with reduced air-speed and open throttle, which is a climbing condition, than at maximum speed in level flight.

The wind-tunnel is very valuable and it is difficult to imagine where we would find ourselves today in aeronautical engineering if no tunnel tests of complete models had ever been made. Certainly our knowledge of stability, as well as our ability to predict performance, would be on a far lower level than they have actually attained. Entirely aside from any discussion as to the theoretical merits of wind-tunnel testing for performance, however, it is evident from practical considerations that such tests cannot be made in all cases. Lack of time and money often forbids the making and testing of a wind-tunnel model, just as it often prevents the carrying out of elaborate performance calculations.

When a new type of airplane is to be designed and when the engineer is confronted with the necessity of making a preliminary choice of its general dimensions and specification, or when there is put up to him suddenly the question of the performance attainable with a given load and power, there is no opportunity to refer the problem to the wind-tunnel or to cover reams of paper with tabulations of resistance. Some shorter method becomes necessary and that shorter method can be found, assuming that the designer's experience with similar types is not sufficient to enable him to guess the performance offhand, only in the derivation and use of simplified performance formulas or of graphs that represent formulas too complex to be readily expressed in analytical form.

Classification of Performance Formulas

Performance formulas may be divided into three principal groups, (*a*) those based solely on experience with airplanes previously constructed, (*b*) those based on rational considerations alone, or perhaps combined with the data drawn from wind-tunnel tests, and (*c*) those which depend on a combination of theory and practice. Up to the present time, the first-named class of formulas has held the premier place and has often proved of great value. These formulas have dealt with the maximum and minimum speeds, with the rate of climb and with ceiling and with every other conceivable element of performance. They have been no less diverse in their origins than in their purposes. Nevertheless, it is difficult to fit a purely empirical chart or formula to the solution of a problem so complex as that of the flight of an airplane in such a way as to be sure that allowance is made for all the variables that may affect the solution. Only by the use of a modicum of theory can we feel certain that we have omitted from consideration in our formulas no variables that should be included and have included none that are unimportant. This does not imply that pure theory furnishes a satisfactory basis for the derivation of formulas, as it cannot do so in the present state of our knowledge. It is always necessary to depend at some point of the reasoning on experience either in the wind-tunnel or in free flight, but theory has great use in guiding our research and of reducing the probability of going astray in the quest for data. The acid test of any formula is application to existing machines for which the true performances have been determined.

The nature of the formulas that can be secured by a combination of rational derivation and of experimental data can best be illustrated by actually following through the process of deriving such expressions. By taking up the elements of performance one by one and determining first the form into which formulas should be cast and the variables involved, actual numerical expressions are obtained which give, as far as possible, the performance of an airplane in terms of its well-known dimensions and other characteristics.

Minimum and Maximum Speeds

The first and the simplest element of performance to be treated is minimum speed. In virtually all instances the true minimum speed of an airplane in steady flight is determined by the wing-loading and the magnitude of the maximum-lift coefficient. The only exception to this rule is found in the case of airplanes whose reserve of power is so small that the horsepower required to maintain steady horizontal flight at sea level and at the angle of maximum lift is greater than the power available under those conditions, and whose ceiling is so low that it could only be regarded as a freak unsuitable for serious consideration. It should be borne in mind, although it is perhaps hardly necessary to emphasize the fact, that the minimum speed that is calculated from the maximum-lift coefficient and the wing loading is the true minimum, and not the minimum speed that is often reported in performance tests as being obtained by closing the throttle until the airplane is no longer able to maintain level flight. The speed of minimum power is far from being the true minimum speed, the latter being determinable in level flight only by keeping the throttle well open and stalling the machine to such an angle that it is barely able to fly level but staggers along at an angle of attack of 18 deg. or thereabout. Few pilots have ever made a serious attempt actually to reach or to maintain the true minimum speed of flight.

If it be accepted that maximum-lift coefficient and load per unit area are the only critical factors in minimum speed, it should be easy to derive a formula in terms of these quantities. Writing the fundamental equation of lift

$$L = L_c S V^2$$

where

$L =$ the total lift
$S =$ the wing area
$V =$ the speed of flight

the equation can be given a form applicable to this special problem by taking the particular values V_{min} and $L_{c\ max}$ for V and L_c respectively and by substituting W, the weight of the airplane, for L, the lift being equal to the weight in steady horizontal flight. If this be done, the equation of minimum speed takes the form

$$V_{min} = \sqrt{} (W/L_{c\ max}\ S) = \sqrt{} (1/L_{c\ max}) \times \sqrt{} (W/S)$$

It remains then only to assume a value of $L_{c\ max}$. If V is given in miles per hour, the values of $L_{c\ max}$ range from 0.0026 to 0.0045. On substantially all modern wing-sections, however, with the exception of a few thick sections used for cantilever wings only, the value will be found to lie between 0.0029 and 0.0038, the higher values appertaining to thick sections and to those with strongly concave lower surfaces. Solving for $\sqrt{} (1/L_{c\ max})$ we see that

$$\sqrt{} (1/0.0029) = 18.6 \text{ and } \sqrt{} (1/0.0038) = 16.2$$

The formula for minimum speed is thus found to be

$$V_{min} = K_1 \sqrt{} (W/S)$$

where K_1 is to be chosen somewhere between 16.2 and 18.6, depending on the general type of wing-section employed. If nothing is known about the wing-section to be used, a rough but nevertheless useful approximation to the landing speed is given by the formula

$$V_{min} = 17\sqrt{(W/S)}$$

In Fig. 1 curves of minimum speed as a function of W/S are plotted for the maximum, minimum, and mean values of K_1. Claims of landing speeds materially lower than those shown by the curves should be viewed with strong suspicion.

The maximum speed offers more complex problems than the minimum, as not only the weight and area and the properties of the wing-section but also the power and the parasite resistance are involved. In attempting the derivation of a rational expression the most convenient and most reasonable simple assumption is that the angle of attack at maximum speed corresponds to the angle of minimum-drag coefficient of the wing. Wind-tunnel tests and free-flight work unite in showing this assumption to be close to the truth in most cases. A small change in the angle of attack at the maximum speed, however, has little effect, as the drag-coefficient curve is ordinarily very flat in the neighborhood of the minimum drag. If

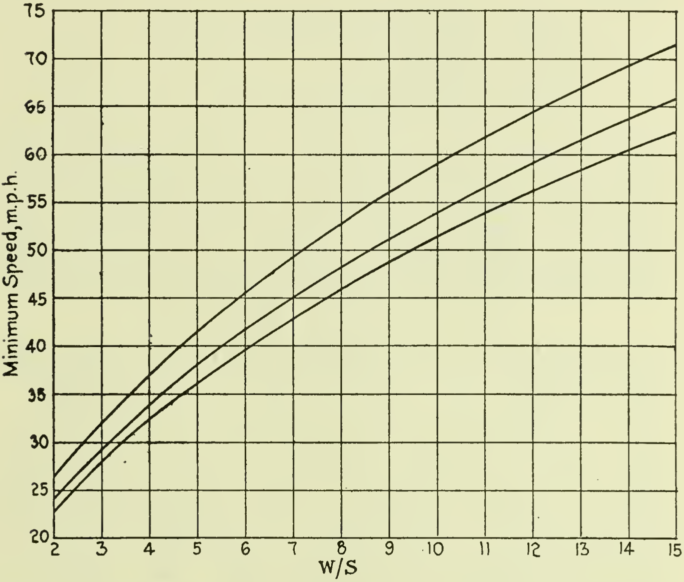

FIG. 1—CURVES OF MINIMUM SPEED AS A FUNCTION OF THE WING LOAD

this assumption be true it is evident that a small change in the angle will have a negligible effect on the drag coefficient, what can then be considered as keeping a constant value. If the drag coefficient of the wings is considered as remaining constant, the coefficient of total resistance may also be taken as a constant, since the change of the parasite resistance coefficient with the angle of attack is obviously small. The equation of the power required at maximum speed may then be written

$$P = (D_cS + R_cs)\ (V^3_{max}/375)$$

where

D_c = the coefficient of wing-drag
R_c = the coefficient of parasite resistance
S = the wing area
s = the parasite resistance area
V_{max} = the maximum horizontal speed in miles per hour

both of these being assumed to remain constant. The power required for level flight at the maximum speed is, of course, equal to the power available, which is the product of engine power and propeller efficiency. Taking the propeller efficiency, which is denoted by η, also as invariable, the equation can be transposed into the form

$$V_{max} = \sqrt[3]{}[(P_o\eta) \div (D_cS + R_cs)]$$

If the ratio of parasite-resistance surface to wing area then be taken as a constant, that is, if the fineness be assumed the same in all cases, the equation becomes

$$V_{max} = \sqrt[3]{}[375\eta \div (D_c + R_c[s/S])] \times \sqrt[3]{}(P_o/S) = K_2\sqrt[3]{}(P_o/S)$$

The rather surprising conclusion is thus reached that, given a certain fineness of design, the maximum speed of an airplane is independent of its weight, provided the machine flies at its maximum speed very close to the angle of minimum drag of the wing. This condition, as already noted, is realized in high-speed airplanes and the few comparative tests made on such machines with different loadings actually show the speed at sea level to be practically unaffected by small changes in weight.

It would, of course, be possible to find the value of K_2, as that of K_1 was determined, by assuming values of D_c, R_c, and s/S and solving directly for the constant, but it is easier and more accurate to call in the assistance of free-flight tests at this point, plotting the values of P/S and of V_{max} against each other in such a way as to determine the value of the constant. This can be done most conveniently by plotting V_{max} against P/S on logarithmic paper, as the curve obtained in that way serves not

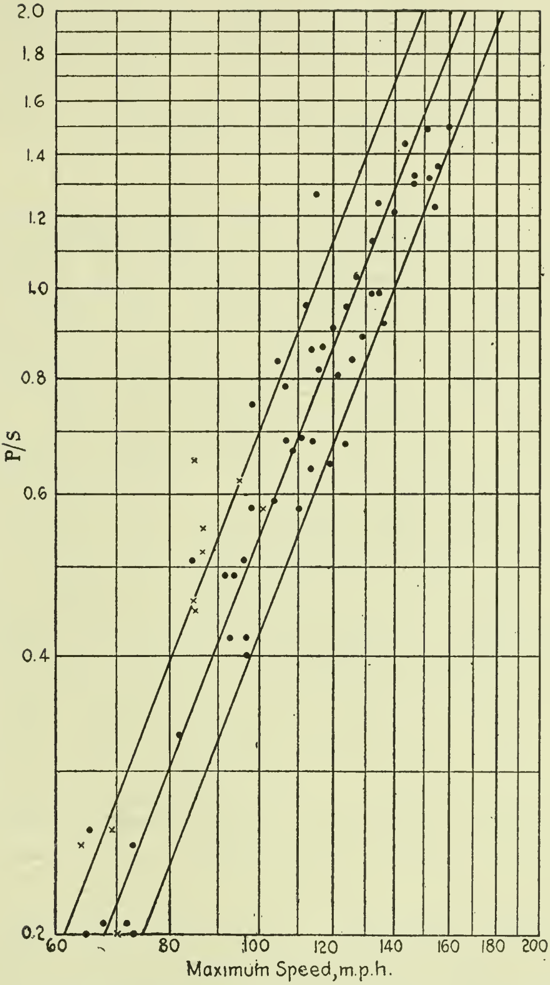

FIG. 2—CURVES SHOWING THE CHARACTERISTICS OF 65 MACHINES AS REGARDS THE RELATION BETWEEN THE POWER AND MAXIMUM SPEED

only to determine the value of K_2 but also to check or to disprove the deduction that the speed varies as the cube root of the power-area ratio. The characteristics of 65 machines have been plotted in this way in Fig. 2 and the best mean straight line drawn through the group. It is found that, when the slope of the line on the logarithmic chart is measured, the slope is not truly $\frac{1}{3}$ but 0.39. The best average straight line corresponds to the equation

$$V_{max} = 127\ (P/S)^{0.39}$$

The two lines parallel to that of best average denote speeds 10 per cent higher and 10 per cent lower, respectively, than those given by the formula. The maximum departures from the formula were found to be 12 per cent high and 17 per cent low for land planes, while the speeds of flying-boats, which are denoted by crosses in Fig. 2, uniformly fell below the speeds calculated by the formula, the maximum discrepancy being 23 per cent. This was to be expected, as the fineness of flying-boats is in general inferior to that of land planes.

The average difference between the speeds calculated by the formula and those found on actual test was 6.2 per cent for all the machines and 5.4 per cent for the land planes alone. These errors may seem large, but the chart includes points for airplanes as diverse in their characteristics as the old JN4D, the Air Service Messenger, the commercial Junkers, the Verville-Packard racer, and the Trans-Atlantic NC boat. In view of the extraordinary range of fineness represented among the machines used, the maximum departure from the mean curve is not excessive. The average errors just mentioned are a good measure of the probable error in using the formula or curve if no attempt is made to allow for differences in fineness. The probable error can be much reduced, however, if some attempt is made to estimate the fineness of an airplane and to modify the constant in the formula accordingly. It is obvious by a glance at some airplanes that they have more parasite resistance than the average for their type, while others have correspondingly less. The exercise of a little judgment in this respect materially decreases the amount of error liable to appear.

The reason for the choice of the exponent 0.39 in place of $\frac{1}{3}$ calls for a few words of explanation, as this change may appear to invalidate all the theory that has been given and to reduce the whole matter once more to guesswork and empiricism. The variation of the exponent is, however, very logical and indeed might have been expected in view of the assumptions made. In the first place, the assumption was made that the airplanes treated by the formula were all to fly at maximum speed at the angle of minimum-drag coefficient. While it has been seen that this is substantially in accordance with the facts for pursuit and racing airplanes, it is far from representing the true conditions of flight for commercial and heavily loaded bombing craft. Machines of these types fly at maximum speed at angles higher than that of minimum drag, and the wing-drag coefficient is therefore greater than for the small fast types. It is logical to expect then that a decreasing value of P/S will cut down the maximum speed more rapidly than would be indicated by a theory based on minimum-drag coefficients in all cases. Furthermore, the derivation of a single formula for the maximum speed rested on the implicit assumption of equal fineness for all machines. This, of course, is far from the truth and it is found in general that the best fineness, or the lowest values of $R_c s/D_c S$, occur in fast machines having high values of P/S. This again would indicate the probability of such an increase in the exponent of P/S as has actually been found.

A number of other empirical and semi-empirical formulas have been derived for the purpose of finding maximum speed, and a few may be cited for comparison with that just discussed. Bairstow, for example, gives the expression

$$(V_{max} - 55)^2 = 1445 \ [(40 \div [W/P] - 1)]$$

for a constant wing-loading of 7 lb. per sq. ft., and the report of the British Advisory Committee on load factors for aircraft presents the alternative formula of similar type

$$([V_{max} \div \sqrt{}(W/S)] - 25)^2 = 196([84.8 \div ([W/P]\sqrt{}[W/S])] - 1)$$

This applies only to single-engine land planes, the values of the constants being changed for other types. Comparison of these two formulas shows a remarkable agreement in view of the modification of the constant, the results being the same within 1 per cent for all reasonable power loadings. Both the British formulas give maximum speeds from 7 to 10 per cent higher than that which I have proposed in this paper. The general agreement in the nature of the change of maximum speed with power loading and wing loading is, however, good; the difference between the results from the Advisory Committee's formula and those from the one derived here being nearly uniform throughout as can be seen from Table 1.

TABLE 1—COMPARISON OF AIRPLANE SPEEDS CALCULATED BY DIFFERENT FORMULAS

		Maximum Speed by Formula, m.p.h.		
W/P	W/S	Bairstow	British Advisory Committee	Warner
7	7	137	136	127
8	7	131	130	121
10	7	120	120	111
12	7	113	113	103
15	7	104	104	95
20	7	93	94	85
9	9	..	136	127
6	6	..	132	127

An American aeronautical engineer has recently devised a maximum-speed formula which has not yet been published, and which I am not at liberty to quote, but which appears to give better results than any of the others that have been mentioned.

CLIMBING CALCULATIONS

After minimum and maximum speed, the most interesting application of formulas is in connection with climb calculations, particularly in the determination of the conditions under which an airplane will have a zero ceiling or, in other words, the limiting conditions under which flight is possible.

The first step in the determination of any climb formula must be the securing of information about the dependence of the minimum power required on the characteristics of the airplane, since all climb calculations depend on the amount of power required for horizontal flight. Such a relation can be obtained by making certain approximations as to the speed at which the best climb is to be secured and as to the L/D of the airplane under climbing conditions. If the L/D is known, the resistance in steady flight can obviously be obtained from the formula $R = W \div (L/D)$ and, if the air speed for the best climb be assumed to be given by the relation $V_1 = K_{min}$, the minimum power required for flight is evidently equal to $WKV_{min} \div (375\,L/D)$, V_{min} being given in miles per hour. Substituting the expression already obtained for V_{min}, this formula becomes

$$P_{req} = [KW \times K_1 \times \sqrt{(W/S)}] \div [375(L/D)]$$

The horsepower available is of course equal to $P_0 \eta$, and it is therefore necessary only to choose values for K, K_1, L/D, and η in order that formulas for the approximate solution of all climb problems can be obtained. In particular, it is evident that the limiting condition under which flight is possible is that in which the horsepower available and the horsepower required are equal at sea level, or that in which

$$W/P \times \sqrt{(W/S)} = [375(L/D)\eta] \div K_1 K$$

At the angle of best climb, wind-tunnel tests and such free-flight information as is available alike indicate that the L/D for a complete airplane may be expected to lie in the neighborhood of 7.5. It has already been seen that 17 is the average value of K_1. The propeller efficiency

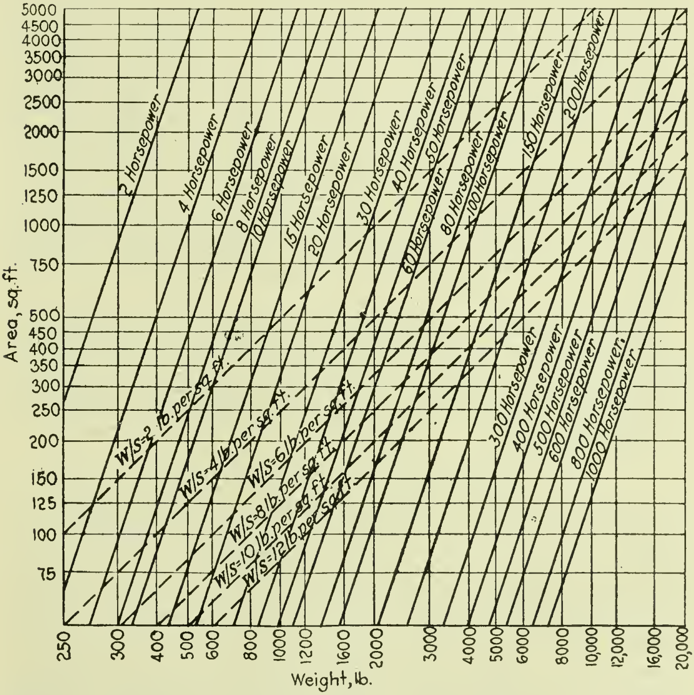

FIG. 3—CURVES OF THE MINIMUM POWER NECESSARY FOR FLIGHT PLOTTED IN TERMS OF THE WEIGHT AND THE AREA

under climbing conditions may be taken as 70 per cent. Finally, experience shows the climbing speed of an airplane to be on the average 35 per cent greater than the minimum speed, and K may thus be taken as 1.35. Combining these factors, the equation for horsepower required becomes

$$P_{req} = 0.0082\, W \times \sqrt{(W/S)}$$

and the limit of possible flight is given by the condition

$$(W/P) \times \sqrt{(W/S)} < 85.9$$

The minimum power necessary for flight in accordance with this formula is plotted in Fig. 3 in terms of weight and area. The full lines in Fig. 3 represent various engine powers, while the dotted lines are loadings in pounds per square foot. The application of the chart is simple. If, for example, it be desired to find the minimum power with which a total load of 3000 lb. can be carried without going to a wing loading of less than 4 lb. per sq. ft., the power is read off at the point corresponding to 750 sq. ft. and 3000 lb. and found to be 69 hp. If a reserve of 25 per cent of the total horsepower is desired as a minimum, it is deduced that the minimum practical power with which such a machine could be taken off the ground would be 69 x 4/3 or 92 hp. Similarly, if we seek to find the maximum load that can be carried by a 100-hp. engine with a wing loading of not less than 6 lb. per sq. ft., the result will be found at the intersection of the full line marked 100 and the dotted line representing 6 lb. per sq. ft. This intersection corresponds to 3500 lb. and 585 sq. ft.

The rate of climb is determined by the familiar method of dividing the excess of power available over the power required by the weight of the machine. The rate of climb in feet per minute at sea level should therefore be given by the formula

$$V_c = 33{,}000\, [(P/W)\eta - 0.0082\sqrt{(W/S)}]$$

In this connection it will be wiser to take 65 per cent rather than 70 per cent as the propeller efficiency in climbing, since the machine that is barely able to fly will have its propeller designed for the one speed at which it can get off the ground, while the airplane with a more normal climb and reserve of power has its propeller designed for a speed nearer the maximum and therefore works more inefficiently under climbing conditions. Taking account of this the expression becomes

$$V_c = [(21{,}400 \div [W/P]) - 271\sqrt{(W/S)}]$$
$$= [271 \div (W/P)] \times [79 - ([W/P] \times \sqrt{[W/S]})]$$

Fig. 4 shows how the rate of climb at sea level should vary as a function of the loading per square foot and the loading per horsepower.

Another performance factor of interest, especially in commercial airplanes, is the angle of climb, which controls the possibility of getting out of a small field over a barrier. The sine of the angle of climb is, of course, proportional to the ratio of the climbing speed to the air speed. The formula for the climbing speed is given

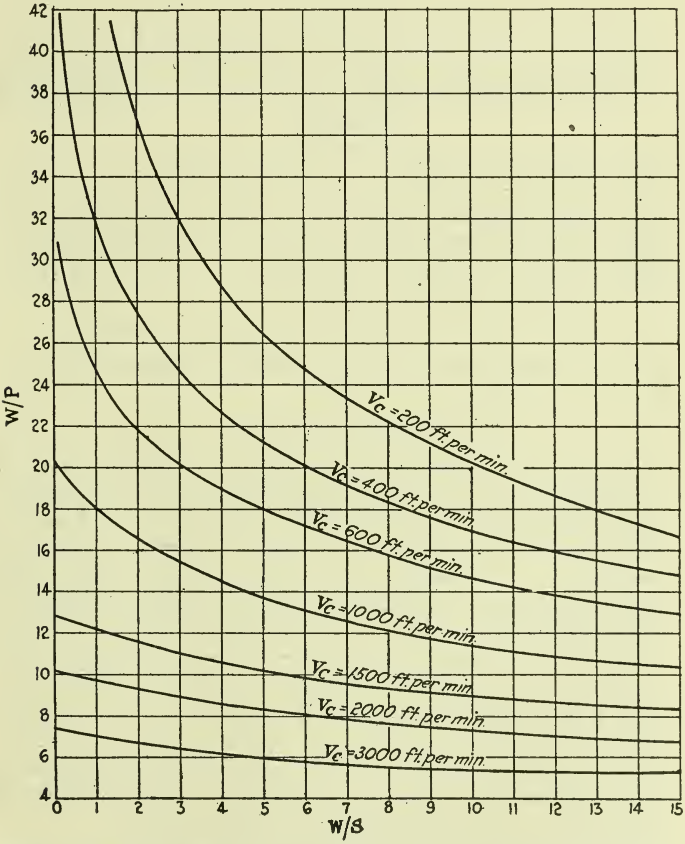

Fig. 4—Curves Showing How the Rate of Climb at Sea Level Should Vary as a Function of the Loading per Square Foot and the Loading per Horsepower

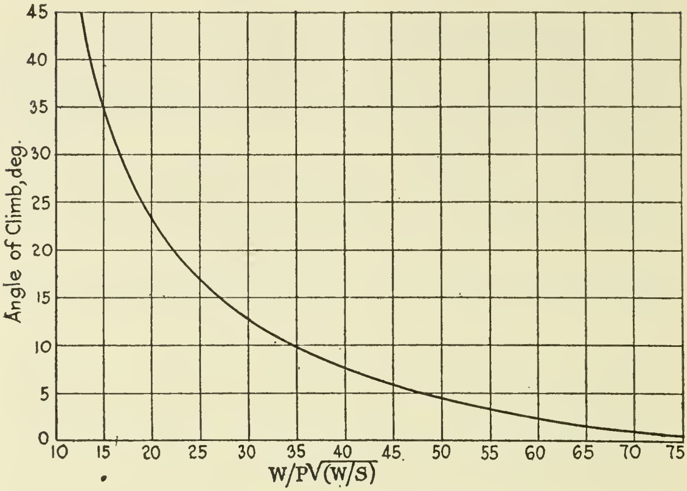

FIG. 5—RELATION BETWEEN THE ANGLE OF CLIMB AND THE WEIGHT

just above and it has been seen that the air speed in climb may be taken as 22.9 $\sqrt{(W/S)}$ in miles per hour, or 2015 $\sqrt{(W/S)}$ in feet per minute. The formula for climbing angle therefore becomes

$$\sin \Theta = (10.62 \div [(W/P) \sqrt{(W/S)}] - 0.135)$$

A curve of Θ against $(W/P) \sqrt{(W/S)}$ is plotted in Fig. 5.

CEILING FORMULA

The determination of the formula for ceiling in terms of weight, area and power alone requires some preliminary analysis in order that the ceiling may be found in terms of the horsepower available and the horsepower required at sea level. It is well known that the minimum horsepower required for flight varies inversely as the square root of the air density, and it has been found by experience that if an exponential relation is required a satisfactory approximation to the variation of the engine power with the altitude is given for most engines by the formula $P = P_o^{1.15}$, P_o being the power at sea level. The maximum power available and the minimum power required must be equal at the ceiling if the propeller is designed so as to give the best ceiling possible.

The horsepower curves at the ceiling with such a propeller would have a common horizontal tangent, as indicated in Fig. 6. The condition of flight at the ceiling would then be expressed by the equation

$$P_{ro}(\rho_h/\rho_o)^{-0.5} = P_{ao}(\rho_h/\rho_o)^{1.15}$$

or

$$(\rho_o/\rho_h)^{1.65} = (P_{ao}/P_{ro})$$

where

P_{ao} = the maximum horsepower available at sea level
P_{ro} = the minimum horsepower required at sea level
ρ_h = the air density at the ceiling
ρ_o = the air density at sea level

Taking the logarithms of both sides of this equation we have

$$1.65 \log(\rho_o/\rho_h) = \log(P_{ao}/P_{ro})$$

The relation between the air density and the altitude under normal temperature-gradient conditions is given with sufficient exactness for most practical purposes by the formula

$$h = 69{,}000 \log_{10}(\rho_o/\rho_h)$$

Substituting in the power-density equation the value for the logarithm of the density ratio obtained from this

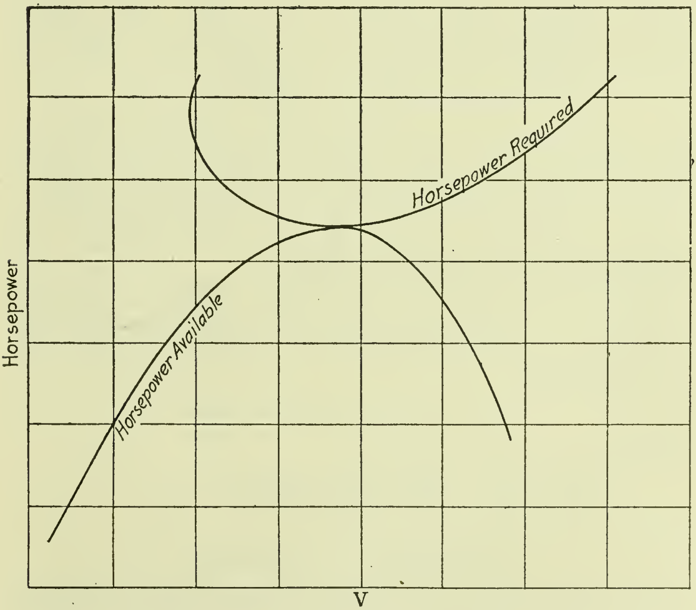

FIG. 6—HORSEPOWER CURVES AT THE CEILING

density-altitude relation the formula for ceiling height becomes

$$H = 42{,}000 \log_{10}(P_{ao}/P_{ro})$$

Referring back to the approximate values given for P_{ao} and P_{ro}, however, it will be seen that their ratio is equal to $85.9 \div (W/P) \sqrt{(W/S)}$ and the theoretical formula for absolute ceiling in terms of the principal characteristics of the airplane therefore appears to be

$$H = 42{,}000 \log_{10} [85.9 \div (W/P) \sqrt{(W/S)}]$$

This formula, like the one for maximum speed, has been put to the test of application in a number of airplanes for which performance data were at hand, and the results are shown in Fig. 7. Semi-logarithmic paper was used in this instance in order that the formula just given might plot as a straight line. The points are so grouped as to leave no doubt that absolute ceiling is primarily a function of $(W/P) \sqrt{(W/S)}$, just as maximum speed has been seen to be primarily a function of P/S. The process of drawing the best average straight line through the points in Fig. 7, however, indicates that the ceiling formula, like that for maximum speed, can be improved by slight changes in the values of the constants, the changes in the present instance being very small. The actual form of the expression giving the best mean results for the 46 airplanes considered proves to be

$$H = 40{,}000 \log_{10} [87.6 \div (W/P) \sqrt{(W/S)}]$$

The average error in predicting the ceiling by this formula was 12.6 per cent. Absolute ceiling is both harder to calculate and harder to measure than maximum speed, and it is therefore natural that the average error in the application of the formula should be somewhat larger for the ceiling than for the velocity.

If the time to climb to a particular altitude is required, it can be found by calculating the absolute ceiling and the rate of climb at sea level and then applying the familiar climb formula

$$t = -(H/V_{co}) \cdot \log_e [1 - h/H)]$$

where

H = the absolute ceiling
t = the time to climb to the height h
V_{co} = the initial rate of climb

Applications of Formulas

In closing, it is of interest to consider a few examples showing how these formulas and curves can be applied

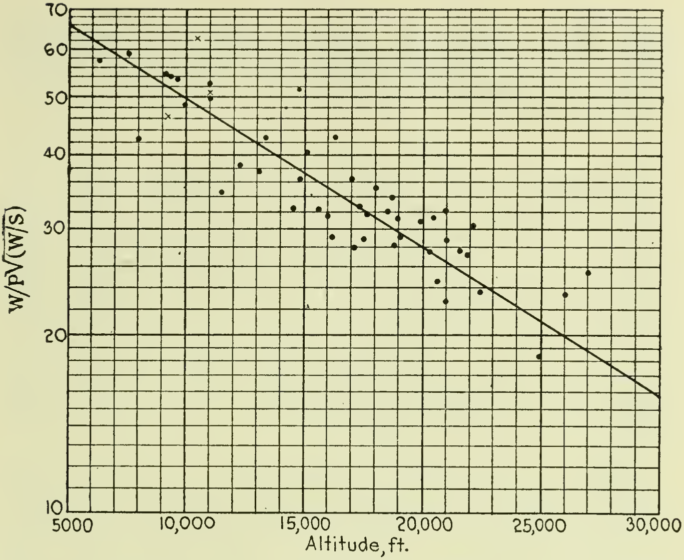

Fig. 7—Application of the Formula for Ceiling to a Number of Airplanes

to specific problems in the quick estimation of the approximate performance of an airplane. Suppose, for example, that it is required to design an airplane to have a maximum speed of 100 m.p.h. to carry 4000 lb. of useful load, including fuel, and to reach an absolute ceiling of at least 12,000 ft. The total weight, the power and the wing area are to be found. The requirement of a speed of 100 m.p.h. gives immediately a power-to-area ratio. By reference to Fig. 2, this is found to be 0.54. A similar reference to the ceiling chart, Fig. 7, shows that an absolute ceiling of 12,000 ft. requires that the value of $(W/P)\sqrt{(W/S)}$ shall not exceed 44. It is now possible to solve directly for the power loading and the wing loading, which are found to be 15.3 lb. per hp. and 8.3 lb. per sq. ft. Assuming the useful load to be 40 per cent of the total weight, the required quantities are found to be 10,000 lb., 655 hp. and 1210 sq. ft. The approximate minimum speed would be 40 m.p.h.

As a second illustration, we may find the minimum wing-area with which it will be possible to carry a total load of 4000 lb. to a height of 20,000 ft., using a 400-hp. engine. The power loading is then 10 lb. per hp., and Fig. 7 shows that $(W/P)\sqrt{(W/S)}$ must not exceed 27.7

if 20,000 ft. is to be reached. W/S therefore must be less than 7.7 lb. per sq. ft., and the wing area must be at least 520 sq. ft. The maximum speed should be 115 m.p.h. and the minimum, 47 m.p.h.

THE DISCUSSION

CHAIRMAN V. E. CLARK:—In connection with Professor Warner's curves on rate of climb and ceiling, he uses the pounds per horsepower times the square root of the pounds per square foot; in other words, the second member of the expression is $\sqrt{(W/S)}$. Did Professor Warner try to introduce into that $(W/S)Ky$ instead of using $\sqrt{(W/S)}$, where Ky is the maximum lift-coefficient of the particular wing-section and the particular arrangement as indicated in wind-drag experiments? If he did try that, did the points come any closer to a common curve?

PROF. E. P. WARNER:—I did not try that out because I was trying for the simplest possible formula for a preliminary prediction. There is no doubt that $(W/S)Ky$ would give a simpler calculation. We find that the machines with thick wings, those having the higher maximum wing coefficient, give a higher ceiling and those with the lower wing-curves give a lower ceiling. That was as close as we could get it without going outside those three factors mentioned. An American engineer has derived a formula for maximum speed that does take this factor into account and appears to give closer results than either this formula or any other that I have ever seen, although I have not checked it up with as many machines as I had here.

H. M. CRANE:—I am not educated to the point of being able to discuss this formula or any of these formulas from the theoretic basis on which they are worked out, but I do want to speak in appreciation of the type of thinking and the type of work that have been done in this paper. It is a very great thing for the average engineering layman to have placed before him in such a simple form the whys and wherefores of airplane performance without going into a long series of calculations. This is the simplest presentation I have seen, and I think that from that point of view it is a very valuable contribution, entirely regardless of its effect on the airplane designer who may use it to make a quick stab at the general proportions of a machine he is working on. It goes much farther than that; it allows the ordinary man who is not doing any of that sort of work to get a fair idea of why one machine does certain things and another machine does something entirely different. I am sure the paper will be of great value to the members of the industry who are interested only casually in airplane designing.

FRANK C. MOCK:—Why does not the parasite resistance cut down the ceiling? If it slows down the plane, it certainly will cut down the climb.

PROFESSOR WARNER:—That is true; it does cut down the ceiling. I did not mean to convey the contrary impression. At minimum speed the wing-drag is much greater in comparison with parasite resistance than it is at maximum speed, though the parasite resistance is apparently of less importance. In maximum speed, the parasite resistance is the big thing. The machines that go to the right of the maximum-speed curve are those in which every effort has been made to eliminate the external bracing resistance. Those that are off to the left of the curve are those that, like the JN-machine, have a large amount of external bracing.

I specified that the exponent used in that curve, if we reduce it to equation form, is 0.39. It is changed to 0.39 empirically, as the result of the plotting of the actual points; because, as a rule, the high-speed machines have a lower parasite resistance than the low-speed machines. Occasionally, however, we get a low-speed machine, such as the Junkers, that has very little parasite resistance. Then the formula gives results that are relatively bad, but even under such conditions the error resulting from the use of the formula is not over 10 per cent.

RECENT AIRPLANE DESIGN AND PERFORMANCE IMPROVEMENTS[1]

By Lieut C N Monteith U S A[2]

The author presents, in outline only, the various features of airplane-development investigation that have been prosecuted. After mentioning the principal types of airplane designed and built shortly before the armistice and the types in service on the battle front at that time, four specific requirements for increasing the speed, the rate of climb and the ability to reach great altitudes are enumerated and commented upon, the further statement being made that an increase in performance can result from any one or from a combination of all four.

Remarks upon design features are interspersed with the discussion of performance improvements, brief explanations being given of the variable-area and the variable-camber-wing schemes, the idea of having a thick wing-section with trailing and leading edges hinged, and that of modifying the wing-section by making the leading edge a small detachable airfoil that can be shifted. All-metal airplane construction is considered under four specific headings in conclusion.

In presenting this subject briefly, it will be possible only to outline the various lines of investigation that have been carried on. Many of the improvements brought to the attention of the public within the last 2 years have been the results of work actually accomplished just prior to the armistice. Necessity for economy following the war has slowed up aeronautical research to an appreciable extent, and the production of airplanes in large quantities virtually has ceased. The Gordon Bennett race in France in September 1920 brought out several representative airplanes. With one or two notable exceptions, these were types that were under construction in the respective countries at the close of the war; every airplane entered in this race was a standard pursuit craft, but each had refinements made upon it for the race with the single idea of increasing its speed, which already was high. For instance, the Nieuport that won the race was a stock airplane with "clipped" wings; that is, the wings had less area than those built for the standard airplane. The American Army racer had been adapted to take a 600-hp. Packard engine in place of a 300-hp. Wright-Hispano engine.

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[1] Dayton Section paper.

[2] Chief of airplane design section, engineering division, Air Service, McCook Field, Dayton, Ohio.

It can be said that most of the airplanes that have been attracting attention were designed and built shortly before the armistice. This includes machines such as the Martinsyde Semi-Quaver, the English Nieuport Night Hawk, the French Nieuport No. 29, the Spad Herbemont, the Borel, the Thomas-Morse MB-3 and the Verville VCP. These airplanes have developed high speed, varying from 150 to 175 m.p.h., while those in service at the front at the time of the armistice, such as the English Sopwith Snipe, the French Spad XIII, the Italian SVA and the Fokker D-VII, showed high speeds of from 125 to 145 m.p.h.

PERFORMANCE

An increase in performance, meaning speed, rate of climb and ability to reach great altitudes, can result from any one or from a combination of all of the following four sources:

(1) Engines of high power and lower weight per horsepower

(2) Reduction of parasite resistance, or the power required to pull exposed struts, wires, wheels, and the like, through the air

(3) An increase of speed-range; that is, the ratio of extreme high speed to the landing speed

(4) A decrease in structural weight for carrying the same load with the same factor of safety

The increased performance of the new airplanes mentioned above can be traced directly to the development of higher-powered engines of low weight. A comparison of the Spad XIII with the Thomas-Morse MB-3, for instance, shows that the number of wires and struts and other parasitic resistances is proportionately the same for each. In the airplanes in use on the front at the time of the armistice the power of the engines used averaged about 230 hp. With the development of the Wright-Hispano engine, weighing 632 lb. when dry and developing about 325 hp. at 1800 r.p.m., a large increase in power with a small increase in weight was available. High speed varies directly as the cube root of the ratio of power available, other things being equal. The average speed for pursuit machines at the time of the armistice being 135 m.p.h. for 230 hp., and the speed of the new airplanes averaging 150 m.p.h. for 325 hp., it will be seen readily that this relation holds and demonstrates that the increase in engine power is largely responsible for the increase in performance.

DESIGN FEATURES

Some German designers were endeavoring to increase the performance of their airplanes by the elimination of parasite resistance, making the wings thick enough to brace them internally, and approximately as efficient as the thin wings. Professor Junkers and Herr Fokker did a great amount of work on this. The internally braced wing appeared in the Fokker triplane late in 1917, followed later by the Fokker types D-VI and D-VII biplanes. These were almost identical airplanes but had different engines. Doctor Junkers went further and developed a single-seater fighting airplane similar to his passenger-carrying machine which is so well known in this Country, but only four of these were in commission at the time of the armistice, and they had inconsiderable service at the front.

A comparison of four airplanes will demonstrate clearly what is meant by the elimination of parasite resistance. Let us consider in the order stated the Curtiss JN-4D, the DH-4, the Fokker biplane and the Junkers monoplane. The German experimenters have done much work on thick high-lift wing-sections, adapted to internally braced wings, but the English and French apparently do not favor this type of design. Their latest designs adhere closely to the conventional biplane having the strut and wire interplane bracing. The French designers have not, it seems, departed very much from standard practice, although some of their new designs vary somewhat from the standard arrangement with respect to struts and wires. American designers have not adhered to any one style, and the thick internally braced wing, the well known thin wings that require external struts and wires, and a combination of both, are appearing in new designs.

In connection with the internally braced wing, two very good sections have been developed during the last year by V. E. Clark, now chief engineer of the aeronautical department of the General Motors Corporation. The RAF-15, a thin section developed by the English,

TABLE 1—COMPARISON OF THE STANDARD THIN WING-SECTION WITH TWO THICK WING-SECTIONS

	Wing-Section		
	RAF-15	USA-27	USA-27-C
Efficiency at High Speeds	100	72	87.5
Efficiency at Cruising Speeds	100	81	86.0
Efficiency at Climbing Speeds	100	93	95.0
Front-Spar Depth	100	152	220.0
Rear-Spar Depth	100	184	256.0

has been for some time the best section for ordinary thin-wing construction, and is generally used as a basis for comparison. The two sections designed by V. E. Clark, known as the USA-27 and the USA-27-C, compare very favorably with the RAF-15, as is shown approximately in Table 1, where the RAF-15 is the standard.

Both of these sections are suitable for internal bracing, and are being used now by various designers on airplanes that give promise of being superior to any yet produced for the military or commercial purposes for which they are designed. The Cloudster, an airplane designed and built by the Davis-Douglas Co., of Los Angeles, Cal., has one Liberty engine and uses the USA-27 wing section. It is expected to carry fuel enough for a continuous transcontinental flight.

The development of the thick wing has made possible the design of large monoplanes in which the wings are thick enough to house the engines, thus reducing further the parasite resistance on airplanes mounting more than one engine. A recent German commercial airplane is a large thick-wing monoplane with four engines mounted in the leading edge of the wing.

The third method of improving performance is the increase in the speed-range. For airplanes that have a constant area and a fixed wing-section, the ratio of high to landing speed rarely exceeds 3 to 1, and in most cases it is nearer 2½ to 1; that is, if the speed range is 3 to 1 and the landing speed is 50 m.p.h., the high speed will be generally not more than 150 m.p.h. Two ideas for increasing the speed-range have been worked on for some time, but there has been no practical solution of either until recently. One is the variable-area wing; the other is the variable-camber wing.

VARIABLE-AREA AND VARIABLE-CAMBER WINGS

The most notable of the variable-area-wing airplanes has been developed by a French designer. The wing is constructed so that large flat plates are carried under the leading and trailing edges, forming the bottom of the wing. These plates are extended when taking-off or landing, forming a wing of considerably increased area. The normal area is doubled in this particular airplane; consequently it supports the same weight at a much lower speed. The plates are drawn in for high speed and form a very efficient wing of small area. Results of complete trials on this airplane are not available, but it is understood that it was not entirely successful due to trouble with the operating mechanism. A satisfactory speed-range of from 37 to 125 m.p.h. was obtained; a ratio of 3.4 to 1.

Many so-called designers submit ideas for variable-area wings but, before building an airplane according to their ideas, it is necessary to determine whether the actual gain in lift afforded by their devices more than offsets the additional weight and complication of the apparatus necessary to control it safely. The idea of the variable-camber wing consists in changing the section of the wing so that it can be made to give a high lift for take-off or landing, and then change to a section that is very efficient at small angles of incidence for high speed. The matter of the extra weight of the operating mechanism versus the increased lift is again the determining factor in the feasibility of the idea. The most successful of the designs so far proposed is the racer built by the Dayton-Wright company for the Gordon Bennett race. In this case the leading and trailing edges of the wing were hinged so that they could be lowered for landing or taking-off, thus forming a high-cambered wing with a high lift-coefficient. They were raised for high speed so that the wing was comparatively flat, and efficient at high speed; also, the parasite resistance was decreased for high speed by the use of a retractable landing-gear.

Glenn L. Martin, an American designer, has developed recently a thick wing-section with the trailing edge hinged at a point 70 per cent of the way back on the chord, and the leading edge hinged at 15 per cent. With this combination and with the hinged flap down at a large angle, he has obtained a lift coefficient double that of the RAF-15; and, with the flaps set up at small angles so as to streamline the section, an efficiency is reached at high speed that is about 20 per cent greater than that of the RAF-15. We have every reason to believe that the tests are reliable and, unless there is some structural consideration that will bring the weight of the operating mechanism up to a prohibitive figure, this new section will permit a greatly improved performance.

The Handley Page Company, of England, has done a considerable amount of experimental work with an idea not altogether new but not so well known among designers as variable-area and variable-camber wings. It is to modify the main wing-section by making the leading edge a small detachable airfoil, shifting this small wing a few inches forward of the main wing by using a series of hinged brackets, or fitting it closely in place as the leading edge of the wing, as desired. The small wing is pushed forward for taking-off or landing, which gives the effect of an increase in area and improved lifting power; for high speed the small wing is brought back to its position as leading edge of the main wing, which provides a smaller area in conjunction with an efficient high-speed wing. Several modifications of this idea have been experimented with and flight tests have been made but, to date, we have received no reliable information as to the details of the device. These three devices and their modifications for improving the speed-range are still in the experimental stage, and have not been incorporated as yet in military types.

The fourth method, that of improving airplane performance by a decrease in total weight, has progressed steadily. The decrease in engine weight lies with the engine designer, but the structural weight is not reduced so easily. It requires long and careful research and tests to effect a combination of efficient design and the most suitable materials. The three materials that have been most investigated are plywood, duralumin and alloy steels. Excellent progress on rib and spar design has been made and wing structures averaging 1.0 to 1.2 lb. per sq. ft. are easily attainable, even for internal bracing, but since the increased factors of safety that are required now for military airplanes necessitate the use of heavier structures, the two have practically offset each other.

ALL-METAL CONSTRUCTION

Metal construction in aircraft is not a postwar development. Ever since the first airplanes were built designers have been making attempts to use metal in various parts of their structure. Only recently, however, has the development of the high-strength alloy steels and duralumin made possible the design of an airplane structure that compares favorably, weight for weight, with the wood and wire construction that was formerly employed.

Metal construction has the following distinct advantages:

(1) The designer is assured of an homogeneous material, particularly when steel is used
(2) Metal is not subject to such variations in strength or size with changing atmospheric conditions as wood is; consequently, it is better for working under different climatic conditions. From the standpoint of war materials, metal is better for storage
(3) The use of metallic covering on the wings and the body gives a much longer life to the average airplane, and does not require the care in handling that a fabric covering requires
(4) With efficient design and planning, metal construction for production work can be turned out more cheaply than wood and wire construction

The Fokker type of fuselage, made of welded steel-tubing, and the Breguet fuselage, made of duralumin tubing and steel fittings, were the outstanding departures from wood construction during the war. Toward its close, however, the Junker duralumin construction made its appearance.

Since the close of the war all countries have been investigating the question of metal construction. The Germans have built some large flying-boats and land machines for passenger carrying, using duralumin throughout except for some of the more important members which are made of steel. Fokker, however, is using wooden wings, having plywood covering in his commercial machine. The French designers have not taken up metal-construction development very extensively, but the English have been doing considerable work on it. Their experimental stations have investigated a large variety of steel spars and wing constructions and obtained some structures that appear to be simple, light and effective. The Short company has built an entire biplane, including wings and tail surfaces, of duralumin. Apparently it is a success. The Vickers company has done some extensive work with duralumin, while the Boulton and Paul company is handling a large amount of metal construction. One of the English designers states that he has "reached the conclusion that for airplanes exceeding 2000-lb. gross weight metal construction is superior from every angle." It can be made lighter, stronger and at less cost than wood, and is more durable under all conditions. The design of metal airplanes consists largely of efforts to simplify and reduce to a minimum the operations necessary to obtain the required parts and assemble them.

In the United States metal construction has been confined to experimental work. To date nothing has been done toward putting an all-metal airplane into production. Several airplanes of all-metal construction are now under consideration by the Air Service. They embody entirely new features as regards the construction of ribs, spars and coverings, and give promise of being distinct advances over anything now being built in Europe.

So far as actual construction is concerned, little has been done since the armistice that represents any appreciable advance over the work that was completed up to that time. The experimental stations have, however, made considerable progress in the investigation of the four methods of improving performance, and it is believed that, when the necessity for the strictest economy in Government expenditures shall have been obviated, larger appropriations will be made and the construction of aircraft carried along on a scale that its importance warrants.

THE PRESENT STATUS OF THE AIR-MAIL SERVICE[1]

By Col E H Shaughnessy[2] U S A

The author outlines the history of the Air-Mail Service and states that the recent policy has been to carry out the intent of the Congress, to align the service with the desire of the administration for economy and to discontinue too rapid expansion.

After a description of the routes and divisions and a listing of the present landing-fields and radio stations, the present equipment is outlined and commented upon, tabular and statistical data being presented. The discussion covers the organization and performance of the service, the casualties, the cost of operation and the policy governing future plans.

The Air-Mail Service operated by the Post Office Department was started in May, 1918, the first step being the establishment of a mail route extending from the City of Washington to New York City via Philadelphia, no intermediate stops being made and, at first, Army planes and personnel being used. The planes were of the JN4-H type, equipped with 150-hp. Hispano-Suiza engines, and had a carrying capacity of 150 lb. of letter mail, or approximately 6000 letters. In August 1918 the Post Office Department, having completed a civilian organization, relieved the Army from further duty with the Air-Mail Service, thereupon assuming full charge of all activities and at the same time putting into use the standard E-4 type plane, equipped with the same 150-hp. Hispano-Suiza engine, but having a carrying capacity of 200 lb., or approximately 8000 letters.

The Washington-New York route operated steadily and successfully. This warranted further expansion and in May 1919 the first leg of the transcontinental route was established from Cleveland to Chicago, via Bryan, Ohio, using the DH-4 type plane, equipped with the 400-hp. Liberty engine, having a carrying capacity of 350 lb., or approximately 14,000 letters. Further extension followed rapidly. Service was started in July 1919 from Cleveland to New York City via Bellefonte, Pa.; from Chicago to Omaha, via Iowa City, Iowa, in May 1920; and from Omaha to San Francisco, via North Platte, Neb., Cheyenne, Wyo., Salt Lake City, Utah, and Reno, Nev., in the following September. In addition to the transcontinental trunk-route, additional lateral routes were put into operation from St. Louis to Chicago in August 1920 and from Chicago to Minneapolis in December of the same year. When the present administration took office, there were air-mail routes in operation from New York City to San Francisco, St. Louis to Chicago and Minneapolis to Chicago, over which mail planes were flying regularly during the daylight hours.

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[1] Washington Section paper.

[2] Late second-assistant postmaster-general, Post-Office Department, City of Washington.

Great things had been accomplished in the way of extending the Air-Mail Service, but the situation as we found it was not altogether satisfactory. Criticism was being directed at the service by the public and by the Congress, due principally to the fact that there had been a series of unfortunate accidents during 1920 which had resulted in a considerable loss of life. In addition to this fact, members of the Congress, in committee and on the floor, objected to the manner in which the Air-Mail Service had been extended and financed. The whole matter was given very serious consideration. A careful check indicated that most of the operating difficulties came from too rapid expansion without providing necessary facilities with which to operate efficiently; also, there was need for a thorough understanding with the committees in the Congress.

We decided, first, to carry out the intent of the Congress, since it is not unfriendly to an Air-Mail Service but simply, and very properly, unfriendly to what it thinks is maladministration in any service. Secondly, we determined to align ourselves with the expressed desire of the administration for economy. Thirdly, we planned to put into effect my own thought that we would help aviation much more effectively by stopping the too rapid expansion, which seemed to be making the Air-Mail Service an extra-hazardous undertaking through lack of sufficient facilities, and concentrating our efforts on standardizing and perfecting operation on a more restricted scale.

For these reasons we decided to discontinue the lateral routes. That from the City of Washington to New York City was abandoned in May 1921 and those from St. Louis to Chicago and from Minneapolis to Chicago were discontinued in June 1921. This reduced our expenditures at the rate of $675,000 per annum. Today, we are operating only the New York City to San Francisco route. It is interesting to note that nevertheless our mail planes will fly at least 1,800,000 miles during the fiscal year beginning July 1, 1921, as compared with 1,760,000 miles flown during the preceding fiscal year, this showing being due to increased efficiency in flying performance.

As operated at present, the Air-Mail Service is used as an auxiliary to the fast mail-train service, because we do not attempt to fly at night. We hope that within a reasonable length of time the proposed Bureau of Aeronautics in the Department of Commerce will come into being and start the work of marking the airways for night flying. When this is done, the real value of an air-mail service will become evident at once, for with night flying mail can be carried across the continent in less than 30 hr. As a matter of fact, on a test flight on Feb. 22, 1921, the Air-Mail Service carried mail across from San Francisco to New York City in 25 hr. and 21 min. of actual flying time. This was a notable performance but altogether too hazardous to try again before the way is marked by suitable aids to flying such as lighthouses and beacons.

The transcontinental route is divided into three operating divisions. The Eastern division, from New York City to Chicago, is 770 miles long, with headquarters at the former point. The Central division, from Chicago to Rock Springs, Wyo., is 1125 miles long, with headquarters at Omaha. The Western division, from Rock Springs, Wyo., to San Francisco, is 785 miles long, with headquarters at the Pacific terminal. Air-mail fields are located at Hempstead, N. Y.; Bellefonte, Pa.; Cleveland; Bryan, Ohio; Chicago; at Maywood, Ill. (a checkerboard field); Iowa City, Iowa; Omaha; North Platte, Neb.; Cheyenne, Wyo.; Rawlins, Wyo.; Rock Springs, Wyo.; Salt Lake City; Elko, Nev.; Reno, Nev., and San Francisco. In addition there is an air-mail warehouse at Newark, N. J., and an air-mail repair-depot at Maywood, Ill.

Radio stations are located at the City of Washington headquarters and at all fields except that at Rawlins, Wyo. Navy radio stations are used jointly at Cleveland, Chicago and San Francisco. All the other radio stations are owned and operated by the Post-Office Department.

TABLE 1—PLANES USED FROM MAY 15, 1918 TO OCT. 31, 1921

History	Number
Crashed; Parts Salvaged	81
Crashed and Burned; No Salvage	20
Burned on Ground; No Salvage	7
Withdrawn Unsatisfactory Type; Parts Salvaged	19
Transferred to Army	1
Withdrawn on Account of Age; Parts Salvaged	8
Withdrawn and Stored; Small Type	9
Available Oct. 31, 1921, Flying Condition	50
Available Oct. 31, 1921, now Undergoing Repair	26
Total	221

EQUIPMENT

During the period from May 15, 1918, to Oct. 31, 1921, 3 years and 5½ months, 221 planes have been used in the Air-Mail Service. Table 1 gives their history.

Since June 1921, when the reorganization of the Air-Mail Service took place, great strides have been made in connection with standardizing and improving the flying equipment. From the first a number of different types of plane have been used. These were principally the JN4-H, the Standard E-4, the JL, the twin-engined DH-4 and the single-engined DH-4. We have standardized on the last named, eliminating all other types. This does not mean that the DH-4 is the most suitable plane available, but simply that it is a satisfactory plane for the Air-Mail Service, and that it is good business to standardize on it because the Army has a large surplus of planes of this type, which are transferred to our service, as needed, without expense to us.

In connection with the useful life of the DH-4 planes, based on our experience, Table 2 gives the record of four planes, showing what can be done in this direction. The planes are still in service and in good condition. At present we have 143 planes in the Air-Mail Service; of

TABLE 2—RECORD OF FOUR AIR-MAIL SERVICE PLANES

Plane No.	Date of First Trip	Number of Miles Flown	Number of Hours Flown
71	March 3, 1919	55,416	625
99	Nov. 15, 1919	64,590	699
104	Dec. 18, 1919	58,219	634
175	Oct. 5, 1920	54,079	647

TABLE 3—OPERATING PERFORMANCE OF THE AIR-MAIL SERVICE

Division	Number of Miles Flown	Number of Letters Carried	Scheduled Trips Completed, Per Cent
New York City to Chicago	112,626	3,506,000	98.5
Chicago to Rock Springs	168,805	4,310,760	97.6
Rock Springs to San Francisco	109,587	2,197,520	98.0

these 50 are in serviceable condition, 38 are being overhauled and 55 are in storage awaiting overhauling.

Another interesting item is that during the fiscal year ended June 30, 1921, the Post-Office Department paid $276,109 to companies for rebuilding and converting planes, and purchased eight planes for $200,000. But no planes have been purchased since July 1921 and all conversion work has been done at our own repair-shop at an average cost of $1,200. At the same time our situation has improved materially. We have 14 more planes in serviceable condition today than we had in July 1921.

ORGANIZATION

The Air-Mail Service is directed by a general superintendent located in the City of Washington; he reports to the Second-Assistant Postmaster-General. The field service is divided into three operating divisions under charge of division superintendents located at Hempstead, N. Y., Omaha and San Francisco. There is an assistant superintendent for each division, as well as a local manager at each landing-field. The supply warehouse at Newark, N. J., and the repair depot at Maywood, Ill., are in charge of superintendents who report directly to headquarters. The total number of employes is 479, and the annual payroll is $787,620. Prior to the reorganization on July 1, 1921, there were 521 employes, salaries totaling $864,321.

PERFORMANCE

The operating performance of the Air-Mail Service in recent months has been truly remarkable. During the first quarter of the present fiscal year 98 per cent of all scheduled trips were completed. The record, by divisions, is given in Table 3.

In compiling the figures in Table 3, trips were not counted unless completed. In addition to the 391,018 miles flown with mail on regular schedule, our records show 49,662 miles of test flights and ferry trips, making a grand total of 440,680 miles flown in the 3 months. An exceptionally good record was made on that portion of the route between Cleveland and Chicago, an air-distance of 335 miles. No mail trips were defaulted from April 29 to Oct. 31, 1921, a period of 185 days. During that time 98,890 miles was flown and 3,271,360 letters were carried. Our records show that approximately two-thirds of all mail trips are made in clear weather, the remainder being in fog, rain or snow. During the same quarter there were 11 crashes, 8 of which occurred on air-mail fields and 3 at outside points due to forced landings. Two planes were destroyed entirely but during the quarter there were no fatalities or injuries to employes on regular mail trips. There was, however, one fatality on a ferry trip.

CASUALTIES

During the period from May 15, 1918, to Nov. 30, 1921, 3 years and 6½ months, there were 30 fatal accidents in the Air-Mail Service. The record by fiscal years, from July 1 to June 30, is shown in Table 4.

During 1920, using round figures, there was one fatality for each 100,000 miles flown; since July 1, 1921, we have had one fataility for 800,000 miles flown. In my opinion this very much improved showing is due principally to putting into effect a sensible businesslike program, using trustworthy men and material and leaving the experimenting and stunting to those who are not entrusted with the United States mails. I emphasize the fact that since July 1, 1921, during the time this remarkably fine record was made by our pilots, DH-4 planes

TABLE 4—FATAL ACCIDENTS IN THE AIR-MAIL SERVICE

Year	Number of Months	Number of Pilots	Number of Mechanics	Number of Other Employes
1917	1.5	0	0	0
1918	12.0	2	1	0
1919	12.0	6	1	2
1920	12.0	13	4	0
1921	5.0	1	0	0
Total	42.5	22	6	2

were used exclusively. I cannot say too much in commendation of our air-mail pilots, several of whom have flown 100,000 miles while carrying mail. Since the beginning of the Service, air-mail pilots have flown 3,400,000 miles and carried more than 2,500,000 lb. of mail, or approximately 100,000,000 letters. No finer class of men or pilots can be found anywhere, individually or collectively, than those in the Air-Mail Service. It is a pleasure to recognize publicly the work they are doing. They are worthily carrying on the best traditions of the Service.

Cost of Operation

Statistics have been published regularly showing the operation and maintenance costs of the Air-Mail Service. They are useful for comparative purposes but have not been truly representative, in my opinion, because the figures have not included such items as general overhead expense, permanent improvements, surplus supplies on hand that were not charged out until used, planes purchased that were not charged to the capital account and the like. Using the statistics as they have been published, the purely operating and maintenance cost per mile during the quarter ended Sept. 30, 1920, was 87 cents; for the same quarter of 1921 it was 71⅓ cents, a reduction of 15⅔ cents per mile. We intend to revise the cost sheet so as to include every dollar properly chargeable to the Air-Mail Service. The new method applied to the periods just mentioned would result in a cost of $1.43 per mile for 1920 and $1.04 per mile for 1921, or a reduction of 39 cents per mile operated.

This material reduction in operating cost is chargeable to (*a*) reduced overhead expense, (*b*) increased efficiency of the personnel and (*c*) economy in purchasing supplies, particularly gasoline. We are operating successfully today on domestic aviation gasoline, whereas formerly the so-called fighting grade of gasoline was used. Cost per mile means nothing unless it is translated into ton-mile figures. Taking the last figures quoted, which are truly representative, we find that in 1921 the DH-4 plane, carrying 350 lb. of mail, was operated at the rate of $8 per ton-mile. This year, the same plane carrying the same load costs at the rate of $6 per ton-mile. We have, however, put into service recently an improved DH-4 plane with the load-carrying capacity increased to 800 lb. This makes possible an increase of over 100 per cent in utility with no additional operating cost; bringing the ton-mile cost down to $2.60. No doubt private enterprise can do much better than this, because, as is well known, the Liberty engine is expensive to operate. Our justification for using it is the surplus stock of the War Department, the first cost of it to us being nothing.

POLICY

The Post-Office Department, true to its traditions, wants air transportation of the mail because rapid transit lies in that direction. It is immaterial whether heavier-than-air or lighter-than-air types of carrier are used if the service given is reliable and speedy. However, the Department does not feel that it should operate an air-mail service any more than it should operate steamboat or railroad service after the time when the commercial interests of the Country are ready to step in and take over the burden.

We are willing and eager to cooperate to the fullest extent with those having craft in the air ready to load the mails, in assisting real commercial aviation, but not by any possible chance to enter into contracts with companies or individuals in the prospectus or stock-selling stage who want to use the legend "United States Mails" to sell stock. We do not know how soon the time looked forward to will come, but let us hope that it will come soon, and let us do collectively everything we can do consistently to advance the date. There are important uses for airplanes other than carrying passengers and freight. Our Country still needs the sort of protection that airplanes and airships and airmen can give.

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